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THERMODYNAMICS

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THERMODYNAMICS

BY

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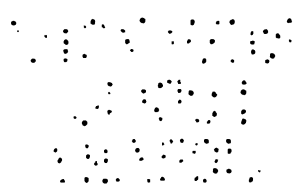

PREFACE

THE subject of Thermodynamics has already been covered by so many able writers that the author has been conscious of a feeling somewhat akin to presumption in the preparation of this book. On the other hand, an experience of several years in teaching Thermodynamics and allied subjects, has convinced him that there is still something to be accomplished in the manner of its presentation. It is a regrettable fact that in too many of our Engineering schools, Thermodynamics is the bugbear of the curriculum. It is approached with dread by the student, and laid aside with a sigh of relief at its conclusion, without his ever having been touched by the real beauty and fascination of the subject.

An attempt is made here to present the subject matter with especial regard to the student's viewpoint. It is the aim in the various developments to proceed progressively so that the reader may easily recognize the relation of each new demonstration that he faces to the whole, and thus avoid that most discouraging of all situations for the student, of working in the dark on what appears to him to be an isolated section of the subject with no connection to the main body.

Two principal features of the book that appear in the endeavor to accomplish the object in view, may be mentioned.

In the first place, the order of presentation of the various subdivisions of the subject is somewhat different from that usually employed in texts on Thermodynamics. Steam is placed first inasmuch as this division of the work always

appears to be easiest for the student to grasp, probably partly because the steam heat engine is more familiar to him as a piece of mechanism; and certainly because the steam heat engine lends itself more easily to study, on the part of the student, because of the fact that its mechanical elements are separated, each one performing but a single function that can be quite well understood. After steam comes naturally vapor refrigeration; then the study of permanent gases, mixtures, and air heat engines. Formal discussion of the rather abstract phases of the subject—the Laws of Thermodynamics and the Kinetic Theory of Heat—is reserved until the student has acquired, in some measure, an experience of his own. It is believed that his better appreciation and understanding of these generalities will far more than offset any temporary need of a knowledge of them in a few places in the book preceding their presentation.

In the second place, wherever possible, the student is presented at the beginning of a section of the subject, with a schematic diagram of an actual heat engine whose principle is to be exemplified. This plan seems to establish a thread of relationship among the several articles that follow, and tends to keep the student anchored to a rather definite conception, and to prevent the development from becoming, for him, a mere abstraction.

A few problems are furnished at the end of several of the chapters. It will be found that these correspond very closely to the matter presented in the chapter. They are arranged in the order of the development, so that in general, it will be found that as soon as a lesson has been covered in a chapter, a problem can be assigned to exemplify the principles. Of course, after all, the problems of the book are chiefly suggestive, inasmuch as every progressive instructor will wish to vary his examples from semester to semester, and also because every different class seems to necessitate problems designed especially for its needs.

The author is deeply indebted to his colleague, Professor

C. H. Fessenden, who has very kindly assisted by offering many valuable suggestions, and by reading proof; to all his associates in the Department of Mechanical Engineering in the University of Michigan for their sympathetic interest in the progress of the work; and to many of his former students whose frank constructive criticisms have served to indicate the value of methods of presentation in the class room, that have been incorporated in this book.

J. E. EMSWILER.

Ann Arbor, Mich.
May, 1921.

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THERMODYNAMICS

CHAPTER I

THERMODYNAMICS AND ENERGY

1. Definition of Thermodynamics.—The Standard Dictionary gives this definition of thermodynamics: "Thermodynamics is that branch of the theory of heat that treats of the relations between heat and mechanical work, especially as acting in a heat engine." The definition at once announces the scope of the subject, and fixes its boundaries.

Two outstanding facts are to be noted:

a. Thermodynamics is a science that deals primarily with energy.

b. Energy appears in many forms. The definition limits consideration to but two of these, viz.. heat and mechanical work.

The energy interchanges that accompany all phenomena of nature are of great interest and importance to scientists in general. But this book is intended to cover only those phases of the subject that deal with the development of energy in relatively large quantities, or upon what may be termed an engineering or commercial scale. Such a treatment of the subject is sometimes alluded to as "Engineering Thermodynamics."

2. Sources of Energy.—In primitive times, man depended upon his own muscles to secure for himself the

things necessary to his existence. Later, he widened his field of conquest by domesticating animals stronger than himself. In the course of time, he invented the water wheel, which made available to him another source of energy. But the real beginning of the epoch of the utilization of energy from natural stores was with the invention of the steam engine, scarcely more than one hundred and fifty years ago. Since that beginning, man's ingenuity has successively perfected energy transformation devices, until today we are calling forth from Nature's stores quantities of energy so vast that serious minded men begin to look with apprehension toward a distant date when those reservoirs shall have become exhausted. Mechanical power has been adapted to one purpose after another so rapidly that what was a luxury or convenience yesterday in everyday life has by its commonness become a necessity today.

Energy is applied in the active or mechanical form. What is the source of active or mechanical energy? With the exception of muscular activity, whose derivation is not yet clearly understood, and which constitutes in magnitude but a small amount of the total energy now employed by civilized peoples, the source of all useful energy is *heat*. Heat appears in the combustion of coal for the production of steam; in the burning of oil or gasoline in the internal combustion engine; and in the use of the gas, which is derived from coal, in the gas burner or the gas engine. When the wind-mill is considered, it is found that the currents of air that furnish the energy are set in motion by the heat of the sun. And even the hydraulic turbine is actuated by water which has been elevated from the surface of the land or ocean by the sun's rays.

3. The Study of Thermodynamics.—The study of thermodynamics, from an engineering standpoint, includes the analysis of the operations that take place in the generation and transmission of power. It is peculiarly a me-

chanical engineering subject, because the mechanical engineer's province lies in that region of engineering which is concerned with mechanisms. The function of a mechanism is to perform an operation or pass through a series of motions. Motion implies an actuating force and a final effect. Thermodynamics is essentially a study of the derivation of operating forces from primary sources of energy.

CHAPTER II

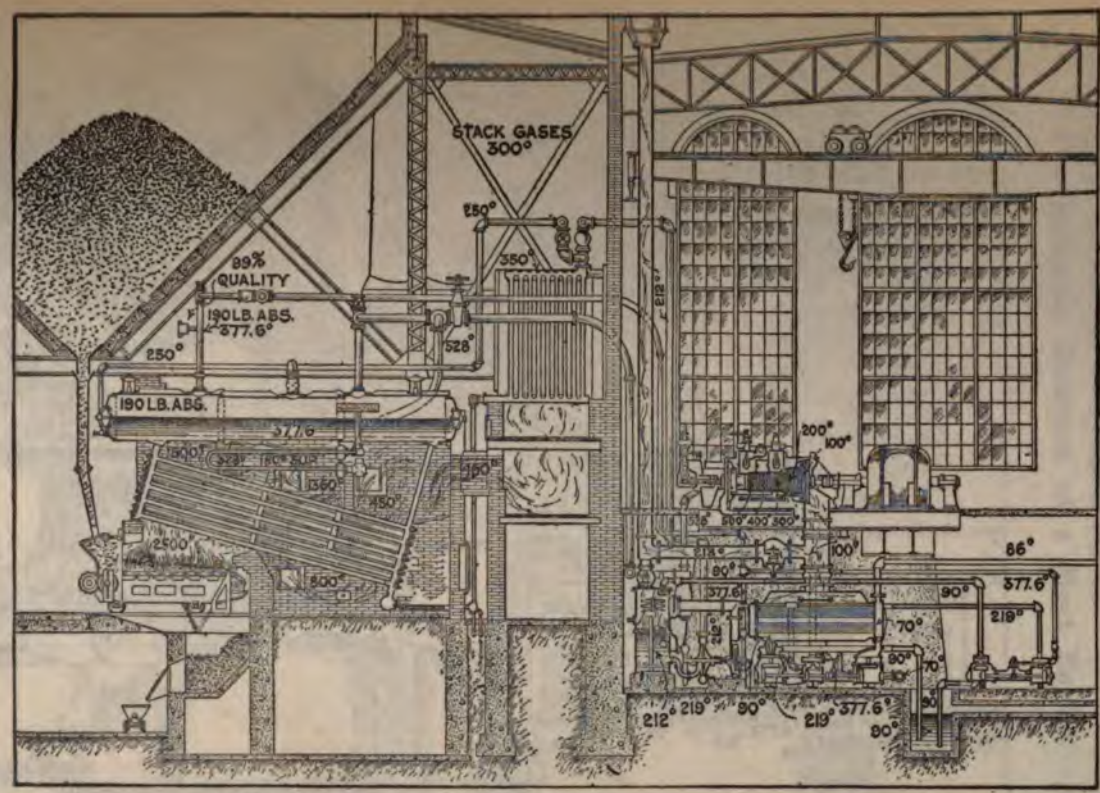
THE STEAM POWER PLANT CONSIDERED AS A HEAT ENGINE

4. The Steam Power Plant.—One of the most familiar combinations of mechanisms employed for the conversion of heat into mechanical work is the Steam Power Plant. The picture of Fig. 1 exhibits the essential apparatus of a typical modern steam power plant. Let it be supposed that the reader steps into this plant with a view to making an extended study of the operations and thermodynamic processes.

The first requirement is that he shall thoroughly familiarize himself with the relation of each piece of apparatus to all the others. He finds the steam, or water, or mixture of the two, to be confined within certain channels, which constitute a circuit or circuits, and upon these circuits are threaded the various pieces of apparatus, each performing its own definite function.

5. Power Plant Diagrams.—For reference and discussion, it is often more convenient to employ a diagrammatic representation of the power plant rather than the picture of Fig. 1. The advantage of the diagram is that it can be sketched at a moment's notice to illustrate or assist in the discussion. Figs. 2 and 3 are examples of diagrammatic representations. To avoid the confusion resulting from too many lines within a limited space, the diagram of the complete plant is separated into two parts—Fig. 2 to represent the main circuit, and Fig. 3 the auxiliary circuit.

6. The Power Plant is a Heat Engine.—The sole purpose of the whole assemblage of mechanisms that con-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300°
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150°
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200°
100°
100°
86°
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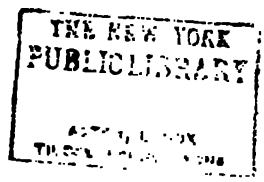
90°

FIG. 1.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stitutes the power plant is to derive mechanical energy from heat. The power plant as a whole is therefore a heat engine, in the thermodynamic sense.

7. The Essential Elements of a Heat Engine.—An

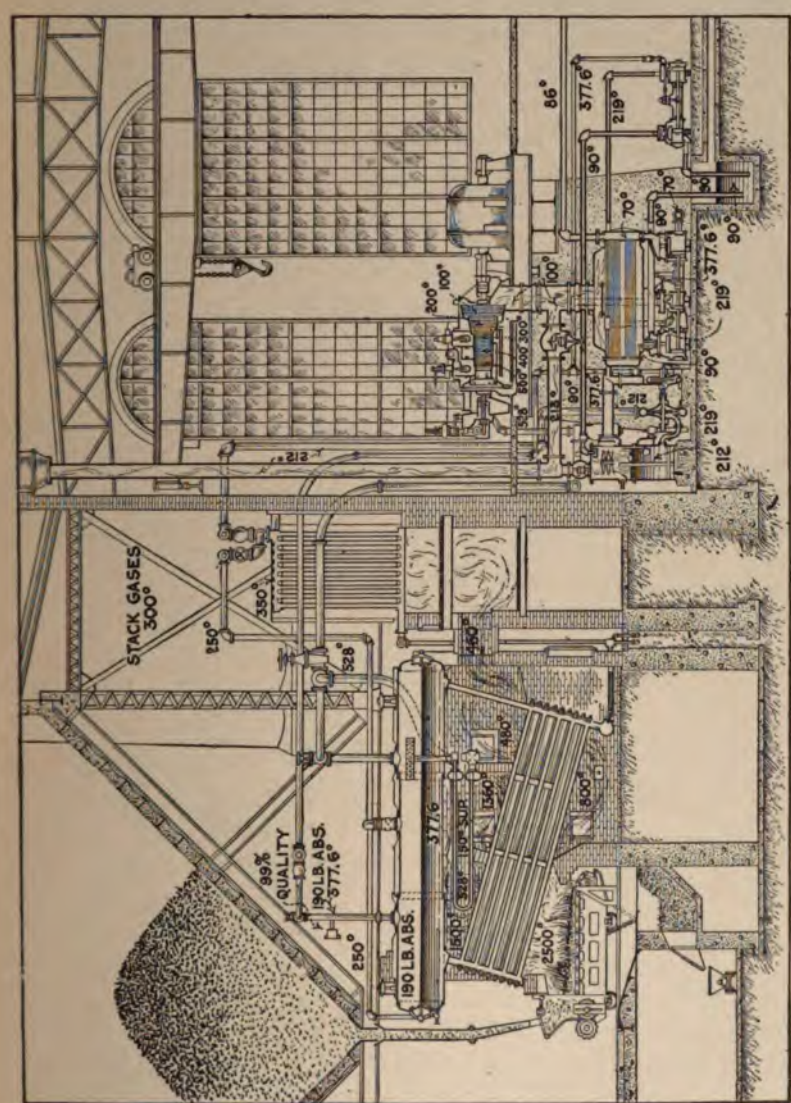

FIG. 1.

inspection of the picture, Fig. 1, reveals the fact that the water or steam, or mixture of the two, receives heat at some points in the circuit and dispenses it at others. It

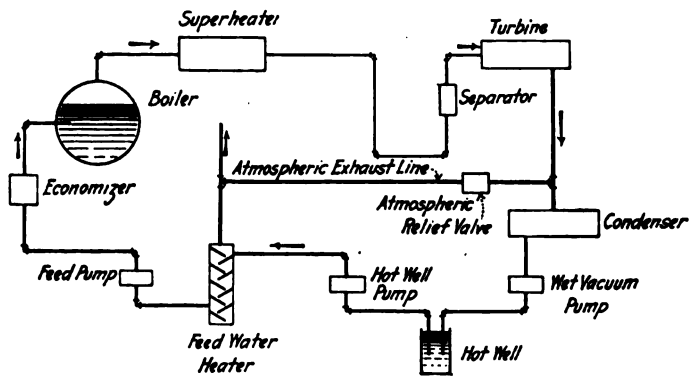

FIG. 2.

constitutes the heat vehicle or *working substance.* The original *source* of the heat supplied to the working substance is in the hot gases resulting from the combustion of the coal in the furnace. Some of the heat is abstracted

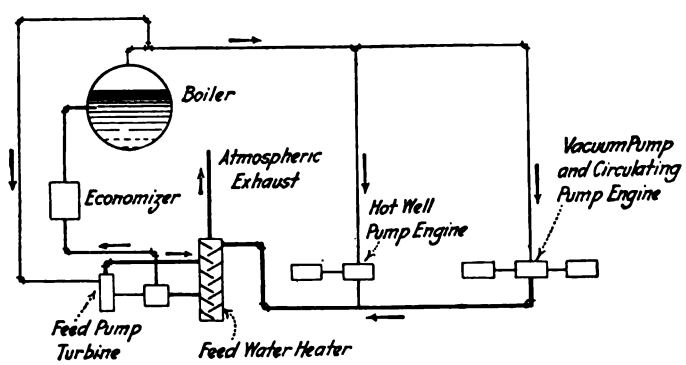

FIG. 3.

from the working substance in the turbine, and converted into useful mechanical effect, which, in the case of the plant illustrated, is immediately changed into electrical energy by the electric generator. The turbine may be

called the *heat utilizer*. It is evident that all of the heat that has been imparted to the working substance at the source is not removed by the heat utilizer, because some is removed in the condenser, and carried away by the cooling water. The condenser is referred to, thermodynamically, as the *refrigerator*. The four essential elements, then, of the heat engine represented by Fig. 1, or indeed of any heat engine, are:

The Working Substance,
The Heat Source,
The Heat Utilizer,
The Refrigerator.

The properties of the Working Substance, steam, will be studied first at some length, and later it will be shown how the steam fulfills its function as an energy vehicle, by making contact with the heat source, the heat utilizer, and the refrigerator, in turn.

CHAPTER III

THE WORKING SUBSTANCE—STEAM

8. The Properties of the Steam.—An investigator who steps into the power plant of Fig. 1 might ask himself how much he knows about the working substance. He would know that at certain points in the circuit the substance is entirely in the liquid state; at other places it is entirely vapor or steam; while at still others, it consists of a mixture of water and steam. He would understand that the pressure is different at different places in the circuit, and likewise that the temperature varies through a wide range. He would further recognize the fact that a pound of the substance would not be of the same size throughout, but would vary in some manner with the pressure and temperature; in other words, he would realize that the specific volume changes as the substance proceeds around its circuit. The fact that the working substance is a vehicle for conveying heat from place to place necessitates the idea of its possessing a different quantity of heat at different points—its heat content varies. These several qualities, viz., pressure, temperature, specific volume, and heat content, which are all related to one another, are spoken of as the *Properties of the Steam*. There are still other properties that have not yet been alluded to, but which will appear as the study proceeds.

9. Observations of the Condition of the Steam.—The investigator will wish to ascertain something definite about the condition of the working substance at the various points of the circuit. Unaided by instruments, he can do but little. He can find out something about the

temperature by the sense of touch. He can learn nothing about the pressure, quality, specific volume, or heat content, except as his judgment relates them to the temperature.

Let him apply gages and thermometers at various points; he is now able to ascertain with accuracy the pressures and temperatures wherever he chooses. The figures marked on the picture in Fig. 1 represent the values observed in this plant. If another instrument, called a calorimeter, is applied at certain points, the *quality* of the steam, that is, the proportion of vapor when the state is a mixture, can be determined. In Fig. 1, a calorimeter is shown attached to the auxiliary steam line leaving the boiler.

10. Other Properties not Directly Observable.—The reading of the instruments alluded to will greatly enlighten an investigator, but how can he determine the other properties referred to in paragraph 8, as well as those alluded to but not yet named, a knowledge of all of which is certainly essential to a complete study of the heat engine? The answer is, the *Steam Tables*. A set of Steam Tables is essential from this point on. All the values used in this book are from the "Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam," by Marks and Davis, 1916 edition.

11. The Steam Tables.—The various properties of steam are all interrelated. If some of them are known by observation or specification, the others may be found by reference to the results of previous extensive physical experimental work on the subject. The Steam Tables present the results of the experimental data obtained by the best physicists who have devoted time to the subject, as interpreted by themselves, or by others who have studied their data.

By the use of pressure gages and thermometers, the pressure and temperature of the working substance at a number of points in the Steam Power Plant of Fig. 1 have

been observed, and noted. Where the steam leaves the superheater the pressure is 190 pounds abs., and the temperature is 528° F. Turning to the Steam Tables, Table II, p. 19, it is found that for 190 pounds pressure, the temperature corresponding is 377.6°. The observed temperature is 528°. The steam leaving the superheater is therefore superheated by an amount equal to 528 − 377.6, or 150.4°. By means of the observation of pressure and temperature, and by the use of the Steam Tables, another property of the steam, viz., the amount of *superheat*, has been ascertained.

Referring again to the Steam Tables, Table III, pp. 52 and 53, it is found that, for 190 pounds, and 150.4° superheat, the specific volume (V) is 3.002 cu. ft. and the heat content (H) is 1281.5 B.T.U.

Thus from the observation of two things concerning the steam, viz., pressure and temperature, the values of other properties can be found by reference to the Steam Tables without the necessity of applying instruments to measure them directly.

Consider another point in the system, where the steam passes directly from the boiler into the auxiliary steam line. The pressure is 190 pounds abs. as before, but the temperature is 377.6°. Reference to Table II, p. 19 (Marks and Davis), as before, shows that there is no superheat since the observed temperature is itself the saturated temperature. Is it correct to read across horizontally with 190 pounds, to determine the other properties designated at the top of the page, as for example, the specific volume of the fourth column? The values of Table II of the Steam Tables are made up for *dry saturated steam* only, that is for 100 per cent quality or 0° superheat. The steam in the auxiliary pipe line may be, and undoubtedly is, at some quality less than 100 per cent; it contains some slight amount of moisture. It is therefore incorrect to read off the properties as they appear in Table II. The observation of pressure and temperature of the steam in the

auxiliary pipe line is inadequate for a complete analysis. It is for this reason that a calorimeter must be applied here, from the indications of which the quality or moisture may be determined. Even with the quality or moisture known, the values of other properties can not be read directly from Table II, nor from any other table, but they can be easily calculated as will appear later. A table might be compiled showing values of properties for each per cent quality or moisture, just as the Superheat Table (Table III) shows the values for various degrees of superheat. Such a table would, however, be very voluminous, and for this reason is not presented.

Consider still another point in the Power Plant, viz., the place in the main circuit where the steam leaves the turbine to enter the condenser. Here the pressure is observed to be .946 pound abs. and 100°. Referring to Table II, p. 16 of the Steam Tables, it is seen that no values are recorded for pressures lower than 1 pound. The reader should therefore turn to Table I, p. 9, where he will find that the temperature observed, corresponds to the pressure, and the steam is saturated. Again the uncertainty as to the quality arises. Unfortunately, no calorimeter is adaptable to the determination of the quality of steam at this low pressure, nor are there any instruments that can be applied to determine by direct observation any of the other properties of the steam in the exhaust pipe. They can be ascertained only by indirect methods, as will appear in later calculations.

In recapitulation, this article brings out the fact that by observation of some of the properties of steam, others can be found from the Steam Tables. In general, it may be said that a knowledge of any two properties is sufficient to determine all the others. The one exception to this statement is the case of saturated steam, where the two quantities, pressure and temperature, are insufficient, and another must be known.

The descriptive matter in the Steam Tables should

be read to gain some acquaintance with the work of the investigators who have furnished the engineer with an instrument of such invaluable aid and marked convenience as the Steam Tables.

12. Saturated Steam and Superheated Steam.— Although the terms "saturated" and "superheated" steam are no doubt fairly well understood by the reader, it will not be amiss to review their meaning here.

Steam is *saturated* so long as it exists in the presence of water. It may be *wet and saturated*, in which case moisture, or water particles are carried along with it, or held in suspension. Wet saturated steam may be constituted of any proportions of water and steam. When steam is cleared of all water, either by complete evaporation, or by mechanical separation, and is not further heated, it is then *dry and saturated*. If the dry and saturated steam is heated further, it becomes *superheated*. Wet steam includes an infinite number of proportions of mixture of steam and water; superheated steam includes all degrees of superheating; but dry saturated steam means only one condition, viz., 100 per cent quality, 0 per cent moisture, and 0° superheat.

For saturated steam, the pressure and temperature are definitely related. For example, at 100 pounds abs. pressure and saturated steam, the temperature is always 327.8° F., no matter whether the quality is 50 per cent or 90 per cent, or 100 per cent. But as soon as the steam is freed of moisture, then its temperature can be raised without raising its pressure.

Problems

1. For a change of one pound per sq. in. in pressure, does the temperature of saturated steam change more in the high pressure region, or in the low? Show by plotting the curve of temperature (as ordinate) against pressure (as abscissa).

2. Show, by curves, how the volume of a pound of dry saturated steam (specific volume) changes with (*a*) the pressure, and (*b*) with the temperature.

3. Show, by a curve, for some particular pressure, for example, 100 pounds per sq. in., how specific volume varies with the degrees of superheat.

CHAPTER IV

PRESSURE-VOLUME AND TEMPERATURE-ENTROPY DIAGRAMS

13. Graphical Representation of the Properties of Steam.—By observation of some of the properties of the steam in the circuit, and by the determination of others from the steam tables, the values of all the properties can now be tabulated for several points in the circuit. The manner of variation of the properties may also be shown graphically to great advantage. Any two of the variables may be chosen as coordinates, as for example, pressure and volume; pressure and temperature; temperature and volume; temperature and entropy; total heat and entropy, etc. Some of these diagrams are found to be of more value than others. The three that are of most use are the pressure-volume, the temperature-entropy, and the heat-entropy diagrams.

The term *entropy* is the name of a property that has not yet been discussed. It is introduced in Art. 17, in connection with the temperature-entropy diagram.

14. The Pressure-volume Diagram.—In Fig. 4, pressures in pounds per sq. in. abs. are represented by vertical, and volumes in cu. ft., by horizontal coordinates. Consider one pound of the steam (or water or mixture). It is proposed to picture the manner of variation of these two properties, viz., pressure and volume, as the pound of substance passes through a complete cycle in the main circuit of the Power Plant.

Begin at the point where the water is about to enter the Feed Water Heater. Its temperature is observed to be 90°, and it is subjected to the barometric pressure, since the heater is open to the atmosphere. What is the volume

occupied by the one pound? Water is practically incompressible, so the volume is not affected by the pressure, although it is influenced by the temperature, and tables showing the variation of the volume, based upon experimental data, can be found in most handbooks. Reference is here made to Table VI in Marks and Davis. At 90° the specific volume is .01610 cu. ft. The pressure at the point of observation in the circuit is atmospheric, or about 14.7 pounds per sq. in. But as indicated above, the effect of pressure on the volume will be wholly neglected. Let the point A, Fig. 4, represent 14.7 pounds per sq. in. pressure and .01610 cu. ft. volume. The volume must be plotted to an exaggerated scale in order to be shown. In fact, the idea here is simply to indicate the manner of variation of pressure and volume without reference to any definite scale.

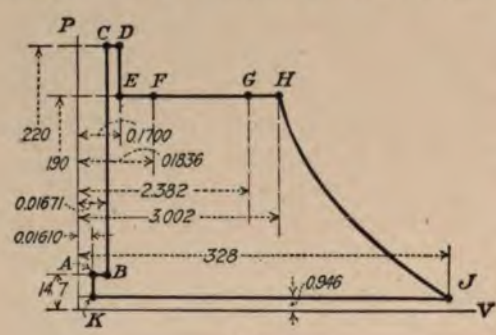

FIG. 4.

In the heater, the temperature of the water is raised to 212°. The specific volume of water at 212° is .01671. The values, 14.7 pounds per sq. in. and .01671 cu. ft. furnish coordinates for point B.

In the feed pump, the pressure is raised to 220 pounds, but the temperature, and therefore the specific volume, is unchanged. For C, the coordinates are, pressure 220 pounds and specific volume .01671.

In the economizer, the temperature is raised to 250° at the feed pump pressure, viz., 220 pounds. The specific volume is .01700, and point D represents the condition after leaving the economizer. The feed water now passes the feed valve, where it is reduced to the pressure of the

boiler, 190 pounds. As there is practically no change of temperature, the volume at E is the same as at D. Upon entering the boiler, the temperature of the water is first increased to 377.6°. The volume of the pound of water, before evaporation begins, is .01836. Point F represents the condition.

Evaporation of the water next takes place at constant pressure and temperature. The volume occupied by the pound of steam as it leaves the boiler to proceed to the superheater, if evaporation is complete, is 2.406 cu. ft. If however, evaporation is not quite complete in the boiler—that is, if the steam contains some moisture, as is usually the case, then the volume is correspondingly less. Suppose a calorimeter is applied to determine the quality of the steam as it leaves the boiler, and by its indication the quality is found to be 99 per cent. The pound of H_2O is therefore composed of 99 parts vapor or steam, and 1 part moisture or water, and the volume of the whole is the sum of the volume of the steam and that of the water, or

$$(.99 \times 2.406) + (.01 \times .01836) = 2.38194 + .00018 = 2.38212.$$

Point G, at 190 pounds and 2.38212 cu. ft., can now be set down in Fig. 4.

It is to be observed that the volume of the water is so slight, as compared with that of the steam (less than 1/100 per cent in this case) that it is negligible. Consequently, in computing the specific volume of wet steam, it is customary and sufficiently accurate to say that it is equal to the specific volume of dry steam multiplied by the quality.

In the superheater, the moisture is first evaporated, and then the further application of heat raises the temperature of the steam, the pressure remaining unchanged. When the steam leaves the superheater, its temperature is 528°, or it is superheated 150.4°. The specific volume is 3.002 from Table III of Marks and Davis, by interpolation. Point H, at 190 pounds, and 3.002 cu. ft. is thus established.

It will be assumed that the steam enters the turbine at the same temperature and pressure as it has upon leaving the superheater, although practically this assumption is not correct.

The state of the steam as it leaves the turbine is not yet known except as to pressure and temperature, and as was shown in Art. 11, these are insufficient to determine the condition completely. In order to complete the discussion, let the quality be assumed for the present, to be $93\frac{1}{2}$ per cent. The values for point J can then be obtained.

The coordinates for the point K, representing the condition of the condensed steam as it leaves the condenser, are .946 pound and .01610 cu. ft. (at 90°). In the vacuum pump, the pressure is elevated to atmospheric, and the cycle is closed at A.

15. The Pressure-volume Diagram Neglecting Water Volumes.—All the changes of volume of the water, as pictured on Fig. 4, are trifling, as compared with the maximum volumes of the diagram. In fact the entire volume of the water itself is so minute that it is entirely negligible in the consideration of the volumes involved in the complete circuit. Disregarding the water volumes, the diagram is drawn as in Fig. 5 and is used in that form.

It is to be noted that the diagram of Fig. 5 (or Fig. 4) is not an *indicator diagram*. An indicator diagram is a picture of the relation between pressures and volumes within the cylinder of a steam engine. The diagrams of Figs. 4 and 5 picture the relations of pressure and volume throughout the cycle of the plant, or the heat engine, in the larger sense.

16. Significance of Area on the P-V Diagram.—During a portion of the time, the steam or working substance is enlarging in size, overcoming resistances, by virtue of which it becomes an agent for doing external work. At other periods it is being compressed or condensed, at which times work must be done upon it. The net result is that it does more work than is done upon it, which is the effect

desired, or ultimate objective of the whole assemblage of mechanisms. The area of the diagram of Fig. 5 is proportional to the net work done by the steam. The area under any line that represents a given portion of the process as, for example, the area $G'GHH'$, under the line GH (which is the line of superheating) shows the work done by the steam because of its growing volume while being heated in the superheater. The area $J'JAO'$ is the work that must be done upon the steam in pushing it into the condenser. The area $AEHJ$ is the net work of the entire cycle, neglecting minor losses.

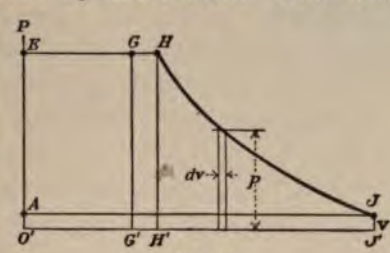

FIG. 5.

The unit of work is the foot-pound, which, in general, is the product of force in pounds times distance in feet; or pressure in pounds per square foot times change of volume in cubic feet. The actual area of the diagram of Fig. 5 even if drawn to scale does not represent work in foot-pounds, because the pressure is expressed in pounds per square inch, for convenience in conforming to the common unit of measurement of pressure.

In the case of the work area, $G'GHH'$, the work done by the steam in the operation is $190 \times 144 \times (3.002 - 2.382) = 16{,}963$ foot-pounds. In the case of the area $H'HJJ'$, the computation is not so simple. The expression

$$\text{Work} = \int P dV$$

must be written, and evaluated by a knowledge of the law of the curve HJ. In some cases, a more direct method can be employed by computing or observing the net loss of heat from H to J.

17. Entropy.—The P-V diagram may be considered as a work diagram. One coordinate, the pressure, represents

what may be called an *intensity* factor; the other, the volume, represents a *distribution* factor. Their product, in general, is work, a form of energy. Both of these factors are readily perceptible to the senses, and there is no difficulty in forming a conception of them.

Heat is also a form of energy, and a compound quantity, the product of two simpler elements. One of these is an *intensity* factor, and the other is a *distribution* factor.

The intensity factor is temperature, which is readily perceptible to the senses, and is capable of direct measurement with instruments that respond to its influence. But temperature alone is no more a measure of heat than pressure alone would be of work. A distribution factor must be conceived which bears a relation to heat similar to that which volume bears to work. This distribution factor is called *entropy*. Entropy, unlike pressure, volume, and temperature, is not a property that is perceptible to the senses, nor is any instrument, yet devised, responsive to its influence.

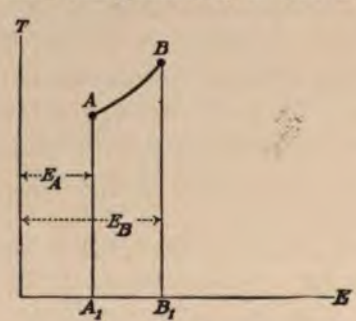

FIG. 6.

18. The Temperature-Entropy Diagram.—In Fig. 6 the ordinates represent temperatures, and the abscissas, entropy. Absolute temperature, degrees F., is used, just as absolute pressure is employed for the work diagram. In the case of entropy, some arbitrary starting point, or zero, must be made use of, since the true zero of entropy is at negative infinity. (See Art. 117.)

In Fig. 6, let AB represent the path of a change. The entropy changes from E_A to E_B. The heat supplied to the working substance is the area under the path of change, viz., A_1ABB_1, just as area on a PV diagram represents work.

19. Representation of the Changes of State of the Steam.—In Fig. 7, it is proposed to picture the manner of variation

of temperature and entropy, and to show the heat changes in a pound of the steam of the main circuit of the Power Plant of Fig. 1, just as the variations in pressure and volume were pictured in Fig. 4 or 5. The unit of entropy has not yet been considered, but this will be deferred for the present. The values for entropy will be taken from the Steam Tables, corresponding to those of the observed properties as marked on Fig. 1.

Start with the water at 90°, the temperature as it is about to enter the feed-water heater. In Fig. 7, the point *A*, representing this initial state, is located at 550° from the zero of ordinates, and at .1114 unit of entropy from the zero of abscissas. This latter figure is obtained from the Marks and Davis Steam Tables, Table 1, p. 9, in correspondence with the temperature of 90° under the column heading " Entropy of Water."

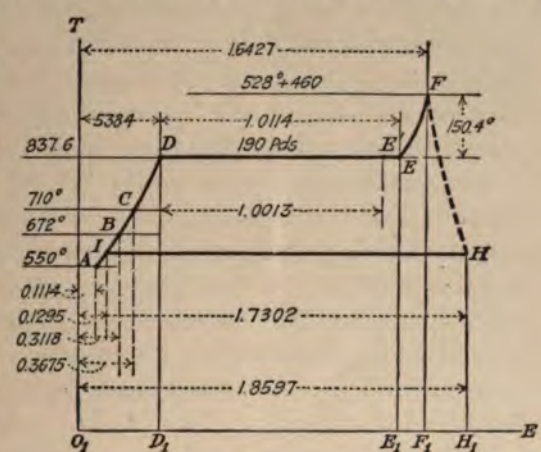

FIG. 7.

In the feed-water heater the temperature is increased to 212° (672° abs.) and the entropy to .3118. *B* represents the state of the substance which is still in the liquid form. The line *AB* pictures the change imposed in the heater. Its direction is upward and to the right because both temperature and entropy have grown, and heat has been supplied to the substance.

In the feed pump the pressure is increased to 220 pounds, but no thermal change takes place; hence the feed pump's activity is not picturable in Fig. 7.

The next operation is that of still further raising the temperature of the water in the economizer; the heating is represented by the line BC, a continuation of AB. After leaving the economizer, the pressure of the water is reduced in the feed valve to that of the boiler, but no thermal change of any importance occurs.

On entering the boiler, the water is first heated to the boiling point D, at 377.6°, and .5384 entropy. Evaporation then begins. Heat is supplied, but with no rise in temperature. The path is to the right horizontally. If the operation continues until the entire pound is evaporated, the state point will be at E. The distance DE is the *entropy of evaporation*, 1.0114, corresponding to 190 pounds pressure. The area D_1DEE_1 is the latent heat of evaporation.

But the steam on leaving the boiler carries with it 1 per cent moisture, as assumed in Art. 14. Only .99 pound of steam has been made, and only .99 of the " Latent Heat " per pound (which is the value given in the Steam Table) has been applied. The entropy change from D is directly proportional to the latent heat applied, consequently the horizontal distance to E', representing the state of steam at 99 per cent quality, as it leaves the boiler, is 99 per cent of the distance DE.

In the superheater, the 1 per cent moisture is first evaporated, this operation being pictured by $E'E$ on the diagram of Fig. 7, after which the steam is superheated, 150.4°. Point F is therefore 988° above the zero of ordinates or 150.4° above the line DE. The entropy is 1.6427 from Table III of the Steam Tables.

By the time the steam has reached the turbine, its pressure has been somewhat reduced by friction, and its temperature, by radiation and conduction; but for simplicity, these minor changes are intentionally neglected.

Within the turbine the pressure drops, the volume increases, work is done, and the steam gives up heat to do the work, because it gets no heat while passing through the turbine. For the present let it be assumed, as was done

in Art. 14, that the quality of the exhaust from the turbine is known to be 93½ per cent. The point H, representing the condition of the exhaust, can now be located thus: the temperature is 100° F. by measurement, which establishes the height of H on the diagram of Fig. 7. If the steam were at 100 per cent quality, its entropy would be .1295 (of water) + 1.8505 (of evap.) = 1.9800. But, since only 93.5 per cent of the pound of substance is steam, it has only .935 of the entropy of evaporation. The total entropy is then $.1295 + (.935 \times 1.8505) = 1.8597$, and point H is located.

The state of the steam before it enters the turbine is at F, and when it leaves the turbine it is at H. A line may be drawn connecting the two points, but it is dotted to show that it is merely a connecting line and not a true state point path. The operation of heat utilization in the turbine will be explained in Arts. 60 and 61.

Following the steam still further, the next operation is condensation, which takes place at a constant temperature. Heat is withdrawn from the substance; hence there is a decrease of area and a decrease of entropy until the steam is all changed to water at 100° and .946 pound pressure. From the tables, the entropy (of water) at 100° is found to be .1295, as already referred to above. The point I falls on the original starting line AB of the diagram, Fig. 7.

Before the condensate leaves the condenser, it is cooled in the liquid state to 90°, where its entropy is .1114, which values are those of the coordinates of the starting point A of the figure. It is true that the substance has not yet completed its circuit, having still to pass the vacuum pump, the hot well, and hot-well pump; but no change of temperature or entropy occurs in these elements. As a consequence, the thermal cycle is really completed when the substance leaves the condenser.

20. The Zero of Heat and Entropy.—Begin with the water at any given temperature, as for example, 90°, as was the case for Fig. 7. Its state is represented by A, Fig. 8,

(which is nearly a reproduction of Fig. 7). The water possesses some heat in this condition. The question now arises, how much heat does it contain? If heat is withdrawn from it, the temperature falls, and the entropy is reduced. If the water is placed in contact with a still colder body, more heat can be withdrawn and the water may be frozen. But ice contains heat because it can be cooled by placing it in contact with a body of still lower temperature. Can any point ever be reached where it can be certain that no more heat can be withdrawn—that the substance has been completely drained of its heat? It is believed that if the temperature of a substance could be lowered to the zero of absolute temperature, it would then be devoid of all heat.

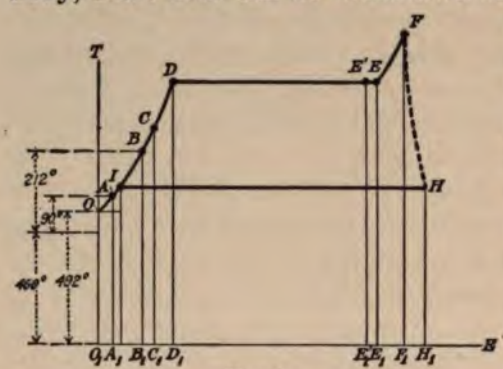

FIG. 8.

It is, however, of small importance just now to know what the absolute amount of heat possessed by any body is. The important thing to know is the amount of heat that is supplied from the source, or that has gone into work; or the amount by which the heat rejected to the refrigerator is less than that which was supplied. All these heat quantities can be determined without any reference to the absolute amount of heat possessed at any time. But it is convenient to establish an arbitrary zero above which to measure heat quantities, just as altitudes are measured above the arbitrary zero of sea level. Water at 32° is said to have zero heat content, and all Steam Tables express heat content above that zero. This particular point of 32°, it should be emphasized, has no special or intrinsic virtue as a beginning place for the measurement of heat, over any other temperature as, for example, 40°, 60°, 100°, etc.

Since the heat of water at 32° is to be considered zero, and since the temperature is finite, it follows that the arbitrary zero of entropy is thus established; or, to express the idea in slightly different form, water at 32° has zero entropy. Therefore, in starting a temperature-entropy diagram for steam or any vapor, a vertical axis is set up above the horizontal base line, and its location establishes the zero of entropy. Then at a point 492° up on the vertical line, the point O (Fig. 8) is marked. The area O_1OAA_1, is the heat that must be applied to the water to raise its temperature from 32° to 90°. The value of this heat quantity is found in Table I, of the Marks and Davis Tables to be 58.0 B.T.U. under the column heading " Heat of the Liquid." This is the heat content, or total heat possessed by the water at 90°, measured above the chosen arbitrary zero of 32°.

21. Structure of a Temperature-entropy.—A diagram like that which has been constructed in Fig. 7 or 8 is of much value in picturing variations of state in terms of temperature and entropy, and in showing heat quantities as areas. But its use can be much extended by making of it a *chart*, which has upon it many lines, with values marked upon them, showing other properties besides those of the coordinates. In Fig. 9, lay off the coordinates for temperature and entropy. In order to assist in understanding the various changes, imagine a tall cylinder, as in Fig. 10, fitted with a frictionless, leakless piston resting upon a pound of water in the bottom of the cylinder. The piston can be loaded by a weight W, such that its mass, together with that of the piston, and the weight of the atmosphere, will produce any desired pressure per square inch upon the fluid beneath. Heat can be supplied by the flame F.

Place a weight W on the piston, sufficient to produce a pressure of 190 pounds per sq. in. abs. on the water. Start with the water at 32°. Point O, Fig. 9, represents the state. Now apply heat; the state point follows a path upward and to the right, because heat is being supplied to the water,

and the temperature increases. The process continues to D, where the temperature of 377.6°, corresponding to 190 pounds pressure, is reached. As heat is further applied, the piston begins to rise, floating on top of the steam as it is formed, but producing the same pressure per square inch. The state point path takes the direction DE, E representing the condition when the process of evaporation is complete. If heat is still further applied to the pound of steam beneath the piston, it will superheat along EF.

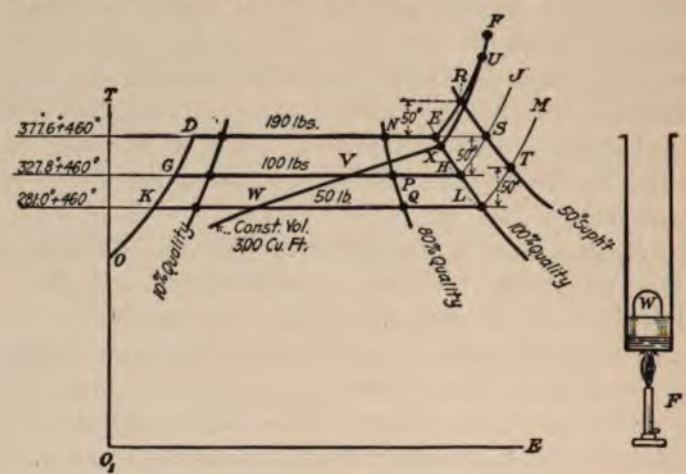

FIG. 9.

FIG. 10.

The one condition or property of the steam that remains constant throughout the entire procedure from O to F, is the pressure. *$ODEF$ is therefore a constant pressure line for* 190 *pounds per sq. in.* It appears in three distinct sections, a different curve for each different state of the substance, viz., all liquid, mixture of liquid and vapor, and all vapor.

Begin with the pound of water at 32° again, but replace the weight on the piston by another, such that the pressure on the water is 100 pounds per sq. in. Apply heat as before and trace the state point path on the plane of Fig. 9. The initial condition is at O, and the state point travels up the

line *OD* as before, because the intensity of pressure on the water has no effect on its rise of temperature. The boiling point of water at 100 pounds per sq. in. pressure is 327.8° F., represented by point *G*, Fig. 9. *GH* is the evaporation line, and it is now necessary to know whether *H*, the 100 per cent quality state, lies to the left or right of *E*, or immediately beneath it. Reference to the Steam Tables shows the total entropy to be greater for the lower pressure, and consequently *H* must be to the right of *E*. The superheat line *HJ* is in general, parallel to *EF*. *The line OGHJ is a constant pressure line for* 100 *pounds per sq. in.*

By changing the weight and repeating the imaginary experiment, another *constant pressure line OKLM* for a pressure of say 50 pounds per sq. in. can be traced. In a similar manner, any number of constant pressure lines can be constructed.

A line drawn through the points *E*, *H*, and *L* furnishes a curve every point on which represents steam at 100 per cent quality. In other words, it is the 100 per cent quality line, or 0° superheat line, or *saturation line*. The line *OD* is the *liquid* or *water line* and also the 0 per cent quality line. If points *N*, *P*, and *Q* are selected on the evaporation lines *DE*, *GH*, and *KL*, respectively, at 80 per cent of the distance from the water line, to the saturation line, the curve that joins these three points is an 80 per cent quality line. In this way, any number of *constant quality* lines can be constructed.

If points *R*, *S*, and *T*, are located on the constant pressure superheat lines *EF*, *HJ*, and *LM*, such that the superheat is 50° in each case, a line drawn through them, will represent a constant value of 50° superheat. It is apparent that a family of such constant superheat lines can be easily constructed for any degrees of superheat desired.

Constant temperature lines are of course horizontal, and *constant entropy lines* are vertical.

Constant volume lines may also be constructed. Suppose it is desired to trace a line any point on which rep-

resents 3.00 cu. ft. specific volume. First, looking up 190 pounds pressure, in Table III, it is found that for 3.00 cu. ft. specific volume, there must be a superheat of 150°, or a temperature of 527.6°, and an entropy of 1.6425, which establishes a point U on the 190-pound line. (Fig. 9.) At 100 pounds pressure, the specific volume of dry steam is 4.43 cu. ft. Evidently, if the pound of H_2O is to occupy but 3 cu. ft., at 100 pounds pressure, it can only be partly steam. Its quality must, therefore, be $\frac{3.00}{4.43} = .677$ or 67.7 per cent, which determines a point V on GH through which the 3.00 cu. ft. line must run. Similarly, for 50 pounds pressure, a 3.00 cu. ft. line will pass through a point W, where the quality is $\frac{3.00}{8.51} = .352$ or 35.2 per cent. Any number of points may be so determined, and the curve $UXVW$ drawn. An entire family of *constant volume lines* may be constructed in this manner.

It is thus apparent that *all* the properties of steam may be pictured upon a plane of which temperature and entropy, or indeed any two properties, are the coordinates. If a sufficient number of lines are drawn to scale, and their values marked upon them, the chart may be used for the solution of problems without resorting to the Steam Tables. However, a chart of this kind, constructed on the temperature-entropy plane, is rarely used. The temperature-entropy diagram serves its best purpose as a means of picturing processes, the operations being indicated by free-hand lines, and the Steam Tables furnishing the quantitative values.

The coordinates that adapt themselves best for chart work, drawn to scale and serving for the solution of quantitative problems, are Heat and Entropy. This chart will be described later (Art. 27).

It is interesting to observe that on a temperature-entropy chart, the liquid and saturation lines approach each other as the temperature is increased. If a sufficiently

high temperature is attained, the two lines will merge into one as illustrated in Fig. 11. This is called the " critical " temperature, and occurs at 706° F. and 3200 pounds per sq. in. for steam. If a weight producing 3200 pounds per sq. in. were placed upon the piston of Fig. 10, resting upon a pound of water, and if heat were then applied, the water would rise in temperature, and finally merge into vapor without appreciably raising the piston. The latent heat would be zero. At the critical temperature, the total heat of the pound of steam would consist only of heat of liquid. The pound of steam would occupy the same space as the pound of water.

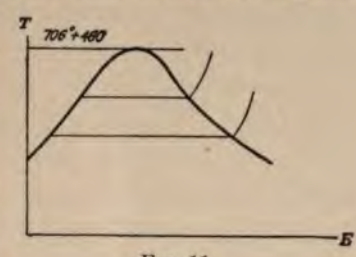

FIG. 11.

22. Values and Representation of Heat Quantities.— Returning now to the power plant of Fig. 1, and beginning with the water at 90°, as it is about to enter the feed-water heater, the first operation is a heating of the water from A at 90°, to B at 212° (Fig. 8). The area O_1OAA_1 is the heat that must be added to bring the water from 32° to 90° and is found from the steam tables, to be 58 B.T.U. The area O_1OBB_1 is the heat that must be added to bring the water from 32° to 212°, and is found to be 180 B.T.U. The difference, $180-58=122$ B.T.U. = area A_1ABB_1 = the heat added in the feed-water heater.

Similarly, the heat supplied in the economizer is found to be the heat content of the water (heat of liquid) at 250° less the similar quantity for 212°; or $218.5-180=38.5$ B.T.U. It is represented by the area B_1BCC_1.

Further heat is applied to the water in the boiler to raise it to the evaporating temperature, represented by the area (Fig. 8), C_1CDD_1. The number of B.T.U. required to do this is found to be $350.4-218.5=131.9$. All of the heat applied up to this point D, measured above 32° is called heat of the liquid (h).

The heat necessary to evaporate the pound of water to 99 per cent quality is the area $D_1DE'E'_1$ and is equal to Lq or $846.9 \times .99 = 838.4$ B.T.U.

In the superheater, the 1 per cent or .01 pound moisture is first evaporated. The amount of heat required to do this is $.01 \times 846.9 = 8.5$ B.T.U. represented by $E'_1E'EE_1$. The total heat H, of the steam at E, viz., at 190 pounds per sq. in. abs. and 100 per cent quality, is a quantity that is given directly in the Steam Tables, as 1197.3 B.T.U., and is represented by the area O_1ODEE_1. It is made up of the sum of the two quantities, heat of liquid, h, and the latent heat, L. Or

$$H = h + L.$$

The total heat of the pound above 32° at the state E', (99 per cent quality), cannot be read directly from the tables, but must be found by a simple computation. It is represented by the area $O_1ODE'E'_1 = O_1ODD_1 + D_1DE'E'_1$, or

$$H = h + qL,$$

where q is the quality of steam at the point considered, and H is the total heat above 32° corresponding thereto.

The superheater then adds the amount of heat $E_1'E'EFF_1$, first evaporating the .01 pound of water and then superheating the steam to 150.4°. The total heat of the steam at F above 32° is read directly from Table III of the Steam Tables (after interpolation) and is found to be 1281.5 B.T.U. It is represented by the area O_1ODEFF_1. The heat of superheat is the area E_1EFF_1 and is obviously equal to the total heat at F less the total heat at E. Or:

Heat of superheat $= 1281.5 - 1197.3 = 84.2$ B.T.U.

After passing through the turbine, the steam is at 100°, .946 pound pressure, and 93.5 per cent quality (the latter by assumption). The state is represented by the point H, Fig. 8. In the condenser, the latent heat of the .935 pound of steam, represented by the area $H_1HII_1 = .935 \times 1035.6 = 968.3$ B.T.U. is first removed. In addition,

the condensate is cooled from 100° to 90°. The heat removed is represented by the area I_1IAA_1, and is found to be 10 B.T.U. by taking the difference between the values of the heat of the liquid at 100° and 90°.

The cycle is now complete and all heat quantities have been considered, except the most important one, which is the heat converted into work in the Heat Utilizer, or in this case, the turbine.

23. Efficiency.—Summarizing the heat quantities that have been supplied at the several points in the circuit:

Heat supplied in feed water heater	122.0	B.T.U.
Heat supplied in economizer	38.5	
Heating of water to evaporation point in boiler	131.9	
Evaporation to 99 per cent quality in boiler	838.4	
Evaporation of 1 per cent moisture in superheater	8.5	
Superheating in superheater	84.2	
Total amount of heat supplied	1223.5	B.T.U.

If it is desired to know only how much heat has been supplied, and not how much is contributed by each of the elements of the plant, the value can be arrived at by simply subtracting the total heat of the substance to start with, viz., 58.0 B.T.U., from the total heat it contains on leaving the superheater, 1281.5 B.T.U. (both quantities being read from the Steam Tables). The result will be 1223.5 B.T.U. as before.

The heat rejected in the condenser is the latent heat, H to I, Fig. 8	968.3	B.T.U.
Cooling of condensate, I to A	10.0	
Total heat rejected	978.3	B.T.U.

The difference, $1223.5 - 978.3 = 245.2$ B.T.U., is the amount of heat disposed of by the turbine. If the small amount of heat lost through the walls of the turbine by "radiation" is excepted, it may be said that the 245.2 B.T.U. have gone into useful effect, into producing work at the shaft of the turbine. The thermal efficiency of the plant—the heat engine—is then

$$\text{Eff.} = \frac{245.2}{1223.5} = .200 = 20.0 \text{ per cent.}$$

Efficiency may be written in two ways, both of which really mean the same thing, but suggest two different modes of arrival.

(1) Efficiency of a heat engine

$$= \frac{\text{Heat supplied} - \text{heat rejected}}{\text{Heat supplied}}$$

(2) Efficiency of a heat engine $= \dfrac{\text{Heat utilized}}{\text{Heat supplied}}$

The first expression was applicable in the case just worked out, but the second will suggest the direct measurement of the useful work developed by a prime mover, in the determination of efficiency.

24. The Expression for Entropy.—In the case of a constant temperature change, of which evaporation is an example, the relation among temperature, entropy and heat quantity is quite simple. In Fig. 12, the line AB represents a change at the constant temperature of T_A. The heat supplied during the operation, A_1ABB_1, is the product of the temperature times the change in entropy. Or

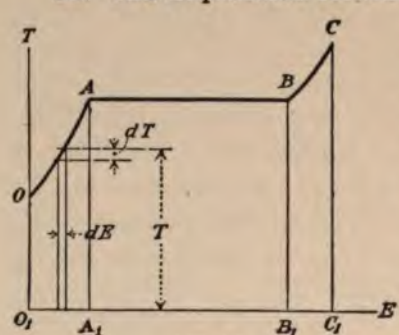

FIG. 12.

$$A_1ABB_1 = T_1 \times (E_B - E_A)$$

When heat is applied, accompanied by a changing of temperature, as in the case of heating water, or superheating steam, the simple relation above does not hold, and an integration of a differential expression must be resorted to. Consider the heating of water along OA. The area of the differential strip is

$$dH = TdE$$

In applying the heat dH, the temperature of the water will

be raised dT degrees. If the specific heat is c, then the amount of heat is given by another expression

$$dH = cdT,$$

whence

$$TdE = cdT$$

$$dE = c\frac{dT}{T}$$

The entire change of entropy from O to A is given by the integration of the last expression between T_O and T_A.

$$E_A - E_O = c\int_{T_O}^{T_A}\frac{dT}{T} = c\log_e\frac{T_A}{T_O}$$

To compute the change of entropy, for a given change of temperature, it is only necessary to know the mean value of the specific heat and the terminal temperatures. Since the specific heat of water is not the same as for superheated steam, it is obvious that the changes of entropy for the same change in temperature are not alike in the case of the two lines OA and BC. In fact, the same thing is true of different sections of the same line, not only because the value of the specific heat is variable, but because the line, even for constant specific heat, has a constantly changing slope.

25. The Unit of Entropy.—In the Pressure-Volume or work diagram the name and value of the work unit, represented by area, are derived from those of the linear coordinates of force and distance; whereas, in the Temperature-Entropy diagram, the unit of temperature (a linear quantity), and that of heat (an area) are first familiar to the reader, and the unit of entropy is expressed or defined in terms of them. Thus the unit of entropy is the amount by which the entropy of a substance is changed by the application of one B.T.U. when the temperature is 1° F. abs. Since there is no definite conception of what entropy is, it is useless to waste time over the significance of the unit. It is customary merely to refer to a change of entropy as so many units of entropy.

26. Specific Heat of Superheated Steam.—The discussion of the expression for entropy in Art. 24 brings up the question of *specific heat*. A full treatment of this property is reserved until the general subject of the permanent gases is considered (Arts. 105 and 132), but a partial study must be made of it at this point.

Since the superheating in the power plant of Fig. 1, is at constant pressure, and represents the practical case for steam, it is necessary to consider only specific heat at constant pressure (c_p). The value of c_p for steam has been found to be somewhat variable with both pressure and temperature. Table III of the steam tables is made up by using values of specific heat of superheated steam from the most authentic sources, and as a rule it is of small concern to know what values of c_p were used, so long as the heat quantities given in Table III are at hand. But for the calculation of entropy, and sometimes for other reasons, it is desirable to know what the specific heat is.

One way to determine the specific heat is to work backward from the values of the table. For example, the total heat of a pound of steam at 100 pounds pressure and 100 per cent quality is 1186.3; while the total heat at the same pressure, but at 50° superheat, is 1213.8. The heat of superheat is 27.5 B.T.U. The mean specific heat is $27.5 \div 50 = .55$. In like manner the mean values of c_p at 100° and 200° superheat at the same pressure are .534 and .516, respectively.

In Marks and Davis, p. 97, is given a chart showing mean values for c_p, at different pressures and temperatures.

The chart is especially useful for demonstrating the manner of the variation of c_p with both pressure and temperature. At low degrees of superheat, while the vapor is still near its liquefaction point, the specific heat decreases rapidly with increase of superheat. With a continued increase of the degree of superheating, the specific heat reaches a minimum value, and then begins to increase. This peculiar influence of the proximity of condensation

point upon specific heat, is one of the things that distinguishes a vapor from a " permanent " gas.

27. The Structure of the Heat Chart.—It has already been pointed out that any two properties of steam may be used as coordinates to represent changes of state. For the Mollier Diagram, or Heat Chart, heat content and entropy are chosen, Fig. 13. The heat content is zero when the entropy is zero, consequently the state of zero entropy and heat content starts at O, with water at 32°. As heat is applied, the entropy increases, the state point changing along the liquid line OA which is similar to that of the temperature-entropy diagram. Suppose evaporation begins at A. As heat is applied, the entropy increases in direct proportion, and the evaporation line AB is straight. During superheating, the entropy continues to increase with the heat although with a slight curvature of the line BC. The line $OABC$ is a line of constant pressure.

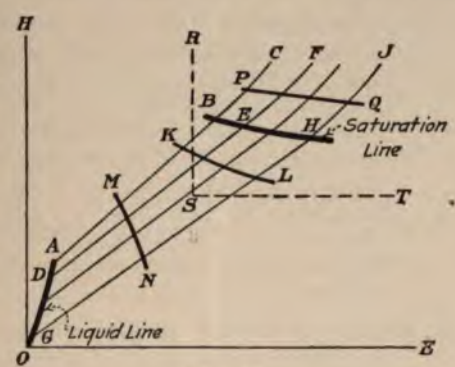

FIG. 13.

Now let other constant pressure lines, such as $ODEF$ and $OGHJ$, be drawn. Through the points B, E, and H, located by plotting the values of total heat and entropy for the 100 per cent quality conditions for the several pressures, the line BH is drawn; it is the line of 100 per cent quality, or the saturation line. The area below BH is known as the region of wet steam; and that above, as the region of superheat. KL and MN are constant quality lines. PQ is a line of constant superheat. Horizontal lines represent constant heat content, and vertical lines constant entropy.

28. The Working Mollier Diagram.—In Fig. 13 imagine the upper right-hand portion of the chart to be cut out by the dotted lines *RST*. The part included within the angle *RST* is the portion represented by Chart No. 1, found in the back of the Steam Tables. It is extremely useful in solving problems where energy quantities are involved.

Problems

1. The following data are obtained from observations of a steam-power plant similar to that represented by Fig. 1.

Boiler pressure	150	pounds per sq. in. abs.
Delivery pressure from feed pump	200	pounds per sq. in. abs.
Exhaust pressure in condenser	1.25	pounds per sq. in. abs.
Temperature of water leaving hot well	80°	
Temperature of water leaving heater	200°	
Temperature of water leaving economizer	230°	
Temperature of steam leaving superheater	480°	
Temperature of condensate leaving condenser	80°	

Quality of steam leaving boiler and entering superheater 98.5%
Quality of steam leaving turbine and entering condenser 95.0%

(*a*) Picture the changes in pressure and volume of the water and steam in a manner similar to Fig. 4 (not to scale).

(*b*) What is the ratio of the volume occupied by the working substance as it leaves the boiler to its volume before evaporation?

(*c*) Neglecting the water volume entirely, picture the changes in pressure and volume of the steam, *to scale*.

(*d*) Picture the changes in temperature and entropy in a manner similar to Fig. 7, *to scale*.

(*e*) Determine the number of heat units supplied to each pound of the working substance in the feed water heater, the economizer, the boiler, and the superheater. Also find the number of heat units rejected to the condenser, and the number taken out by the turbine to be converted into mechanical work (neglecting "radiation" losses). What is the thermal efficiency of the plant?

2. Construct to scale a temperature-entropy diagram showing the following lines:

(*a*) Constant pressure lines for 1, 10, 50, 150 and 500 pounds per sq. in. abs. In the superheat region, determine for each pressure, points at 50°, 100°, 150° and 200° superheat.

(*b*) Draw in the "liquid" and "saturation" lines.

(*c*) Draw in the lines for 95%, 50%, and 10% quality.

(*d*) Draw in the lines to represent constant superheats of 50°, 100°, 150° and 200°.

(*e*) Construct constant volume lines for 2, 5, and 20 cu. ft.

(*f*) Plot points for the liquid and saturation lines as far as the values go in the Steam Tables.

3. By means of curves, and with the help of Table III of the Steam Tables, show how the mean specific heat, at constant pressure, of superheated steam, varies, (*a*) with the degree of superheat for 10, 150, and 300 pounds per sq. in. and (*b*) with the pressure for 20°, 100°, and 200° superheat.

4. Construct to scale, an elementary Heat-entropy chart, similar to Fig. 13, showing the lines called for in Problem 2, above.

CHAPTER V

THE EFFECTS OF HEAT—HOW STEAM DOES WORK

29. Heat and Work.—Although it is understood that the one purpose of the steam-power plant, Fig. 1, is to deliver work energy from heat, and inasmuch as it was shown in Art. 23 that there were 245.2 less heat units rejected from the system into the cooling water of the condenser, than were supplied in the formation of the steam, and that these 245.2 B.T.U. could be accounted for only as work, nevertheless the definite manner whereby heat is converted into work has not yet been shown. It is proposed to give some conception of this transformation in the paragraphs that follow.

30. The Effects of Heat.—Let the tall cylinder and frictionless piston be again used in the illustration of Fig. 14.

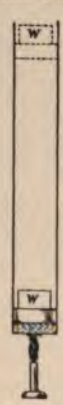

Fig. 14.

Starting with a pound of water at 32° in the bottom of the cylinder, and with a weight W, on the piston, sufficient to produce 190 pounds per sq. in. absolute pressure, let heat be applied by means of the flame. The results of the application are, first, the heating of the water, then the vaporization of the water, and finally the superheating of the steam. Three divisions of the entire heat quantity supplied to the substance are thus indicated, viz., the

Heat of the Liquid
Heat of Vaporization
and Heat of Superheat.

These divisions are marked by the state of the substance, and their significance and application have already been considered in connection with the Steam Tables.

The effects of the heat upon the substance may, however, be classified upon another basis. During the heating operation the temperature of the substance is raised, its state is changed from liquid to gaseous, and at the same time its volume has grown from almost nothing to a relatively enormous magnitude. It takes work to push up the weighted piston, and the energy for doing this work can come only from the heat supplied. Upon this basis, the Heat Effects can be written as follows:

Heat to raise temperature, called *Sensible Heat.*
Heat to change state from liquid to gaseous—called *Internal Latent Heat.*
Heat to do work in raising the piston, or in enlarging the volume of the substance against the resistance of the environing pressure, called *External Work.*

Each of the three heat quantities, Heat of Liquid, Heat of Vaporization, and Heat of Superheat, will now be examined with reference to the disposition of the heat as far as the effects classified above are concerned.

31. Heat Effects during Heating of the Liquid.—In the liquid state, the temperature of the substance is raised from 32° to 377.6 (190 pounds per sq. in. pressure). The heat required for doing this is Sensible Heat, since the effect is observable to the sense of touch or to the eye by means of a thermometer. There is no change of state, consequently the second heat effect is zero in this case. The volume of the water has grown slightly, from .01610 to .01836 cu. ft. The work expended in lifting the piston is $190 \times 144 \times (.01836 - .01610) = 61.83$ foot-pounds or .079 B.T.U. The effects may be summarized thus:

Sensible heat	349.321	B.T.U.
Heat for changing state	0.000	"
Heat equivalent of external work	0.079	"
Heat of liquid	350.400	B.T.U.

The heat equivalent of external work is so small as to be insignificant, and for engineering work it is neglected, in

which case then, all of the Heat of the Liquid is considered as Sensible Heat.

32. Heat Effects During Vaporization.—The heat supplied to effect vaporization is called Latent Heat, because the process is accompanied by no change in temperature. But the entire amount of heat absorbed is not really latent, since the change is accompanied by an enormous growth in volume. The portion of the heat required to do external work in enlarging the volume against the resisting pressure is, in the case of the example, $190 \times 144 \times (2.406 - .01836) =$ 65,325 foot-pounds or 84.0 B.T.U. The distribution of the Heat of Vaporization then is:

Sensible heat	0.0	B.T.U.
Heat for changing state (internal latent heat)	*762.9	"
Heat equivalent of external work	84.0	"
Heat of vaporization (latent heat)	846.9	B.T.U.

The value of the internal energy of evaporation is given in the Steam Tables (Tables I and II), column 9. It is to be regarded as the amount necessary to effect the change of state from liquid to vapor, if no change in volume accompanied the process of vaporization.

33. Heat Effects During Superheating.—Here there is a rise in temperature, accompanied by a growth of volume. Assuming the steam to be superheated to 150.4°, to correspond with the conditions of the Power Plant of Fig. 1, the volume changes from 2.406 to 3.002 cu. ft. The external work done is $190 \times 144 \times (3.002 - 2.406) = 16{,}306$ foot-pounds or 21.0 B.T.U. The entire heat of superheating, found from the Tables, is 84.2 B.T.U., and its distribution among the three effects is:

Sensible heat	63.2
Internal latent heat	0.0
Heat equivalent of external work	21.0
Heat of superheating	84.2

* This quantity does not quite agree with that read from Marks and Davis, because the values there given for the internal energy of evaporation are slightly in error.

34. Summary of Heat Effects.—Summarizing the values from the preceding three paragraphs, the entire heat quantity supplied can be classified as to its effects, as follows:

	Heat of Liquid.		Heat of Vaporization.		Heat of Superheat.		Total Heat.
Sensible heat	350.4	+	0.0	+	63.2	=	413.6
Internal latent heat	0.0	+	762.9	+	0.0	=	762.9
Heat equiv. of ext. work	0.0	+	84.0	+	21.0	=	105.0
Total heat	350.4	+	846.9	+	84.2	=	1281.5

The Sensible Heat and the Internal Latent Heat are alike, in that they both remain in the substance. The heat equivalent of the external work really goes on through the substance and registers its effect upon something external to it. In the case of the tall cylinder illustration, at the end of the process the entire amount of heat taken in by the substance finally appears partly as heat or energy that resides in the substance, and partly as potential energy, external to the substance, in the elevated position of the piston.

It is now evident that the terms Total Heat and Heat Content refer to the quantity of heat taken in by the substance, starting from 32°, and coming up to any specified condition, and not to the amount of heat actually inherently possessed by the steam at any time.

35. How the Steam Changes Heat Into Work—External Work Equivalent.—Let Fig. 15 represent the essential elements of a steam plant. Let the boiler and condenser maintain the same pressures as those of the plant of Fig. 1. The turbine is replaced by a reciprocating engine, in Fig. 15, which is more useful in bringing out the points that immediately follow. The turbine is considered later in Art. 38. For simplicity, there is assumed to be no loss of pressure or temperature in the pipe line from the boiler to the engine.

Begin with the piston at A, its extreme position to the left. With the valve N closed, and the valve M open, if a pound of water is evaporated in the boiler and superheated in the superheater, it will push the column of steam in the pipe before it and move the piston to B. The growth in volume is 3.002 cu. ft. (specific volume of steam at 190 pounds and 150.4° superheat), less .0184 (the volume at the water), and the action, as far as the steam is concerned, is the same as in moving the free piston of Fig. 14. The work done is the external work portion of the heat supplied

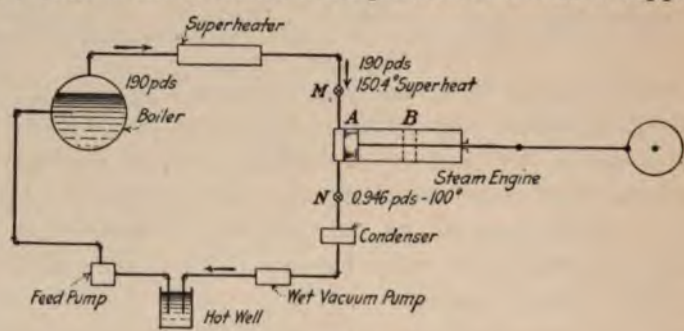

FIG. 15.

to the steam, viz., the equivalent of 105 B.T.U. (Art. 34) and is represented by the area $E'EAA'$, Fig. 16. If the operation of supplying heat at constant pressure could go on indefinitely, in the superheater, and if the piston of the engine could move out indefinitely, work would be continually gained from the heat, and the area $E'EAA'$ enlarged correspondingly. But there are two things that limit this apparently ideal scheme of getting work from heat. The temperature of the steam is going up all the while, and would soon reach an equality with that of the source, when no more heat would flow into the steam; and an engine could not be constructed capable of confining an indefinitely large volume behind its piston. Sooner or later the heat absorbing process must be suspended, the working substance restored to its original state, and the piston of the engine returned to its initial position.

36. Restoration Process and Cyclic Operation.—In Fig. 15 the valve *M* may be closed, and the valve *N* opened, throwing the engine cylinder into communication with the condenser. The consenser is assumed to maintain a pressure of .946 pound per sq. in. abs. All the steam in the cylinder, at 190 pounds pressure, will immediately rush into the condenser, except a cylinder full at .946 pound pressure. In the condensation of the steam, heat must be removed from it. This very important fact is to be noted,—that in the restoration of the steam to its original state, heat must be removed *as heat*. The statement is equally true, whether a surface condenser is considered, in which case the same identical substance is returned to the boiler; or a jet condenser, where the condensed steam is carried away with the condensing water and new water is fed in to replace it; or an engine with atmospheric exhaust, where the steam is dispersed into the atmosphere, held by it for a time, and ultimately condensed and precipitated as rain.

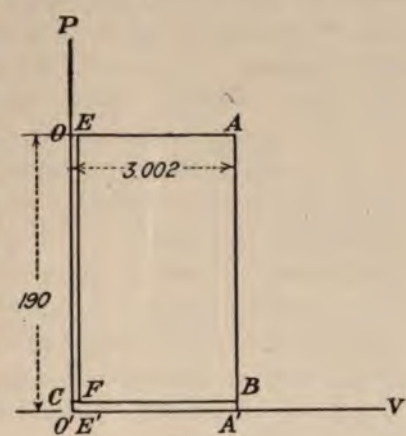

FIG. 16.

In order that the restoration process on the substance may be complete, it must be reintroduced into the boiler. It must have the work of the feed pump done upon it.

In restoring the engine to its initial position (Fig. 15), the piston must push out the steam in front of it into the condenser, and must therefore do work on the steam, in proportion to the back pressure. This operation is represented in Fig. 16 by the line *BF*, and the work of overcoming the back pressure is *A'BFE'*.

The work of the feed pump is the small strip of area *COEF*.

Some of the total work area $O'OAA'$ has been requisitioned to assist in restoring the working substance and engine mechanism to original conditions. Consequently, it cannot all be claimed for useful external purposes. Not all of the 105.0 B.T.U. external work part of the heat supplied (Art. 34), are available for net work effect.

The steam-power plant, viewed as a heat engine, and in fact all heat engines, must operate upon a *cyclic* principle. In the case of the plant of Fig. 15, just discussed, it is true in two senses. The working substance passes through a series of operations, returning always to the same conditions at the same point in its circuit. This is true of all heat engines. No operation can be continued indefinitely upon any working substance. It must pass through a series of changes constituting a *cycle*.

The heat utilizer, or engine, of Fig. 15, also operates upon the cyclic principle as a piece of mechanism, in its relation to the steam. The steam passes in, pushing the piston out, the valves are reversed, and the piston comes back, pushing the steam out. The steam turbine in contradistinction to the reciprocating engine, does not operate upon the cyclic principle, but upon the continuous flow plan.

Let it be clearly understood that the working substance of all heat engines must undergo cyclic changes, but the heat utilizer, commonly called the engine, may or may not operate upon a cycle plan.

37. Possibilities of Useful Work from the Intrinsic Energy.—Thus far only the external work portion of the entire amount of heat supplied has been considered as available for producing useful work effect. Besides the 105.0 B.T.U. heat equivalent of external work, it will be recalled that there are 413.6 B. T. U. Sensible Heat and 762.9 B.T.U. Internal Latent Heat (Art. 34) or a total of 1176.5 B.T.U. of what may be called *Intrinsic Energy*.

It is important to know whether or not any of this large portion of the entire investment of heat can possibly return anything in the way of net useful work.

Refer again to Fig. 15, and consider that the piston has been pushed to B, as a result of the external work effect. The pressure and volume are represented by A, Fig. 17 (190 pounds, and 150.4° superheat) If now, in place of opening valve N, after closing M, Fig. 15, the piston is allowed to move on farther, it is clear that work will be done by the steam on the piston. In Fig. 17, the change of condition of the steam during this expansion is represented by AB. The expansion continues until the piston can move no farther, and this point may find the pressure behind the piston at any value such as B, where the pressure within is just equal to the condenser pressure, in which case, when the valve N is opened, there will be no rush of steam into or out of the cylinder; or it may be at G, where there will be a drop in pressure at release, from G to H; or even at K, lower than the exhaust pressure, so that release is accompanied by steam rushing into the cylinder from the condenser. Let it be assumed that the piston travels to B, in the example that has been employed throughout for illustration.

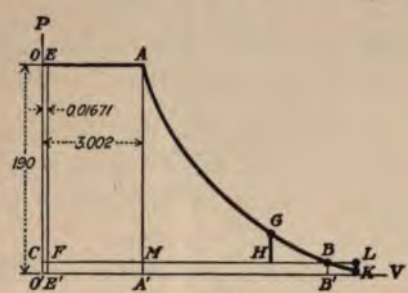

FIG. 17.

The area $A'ABB'$ (Fig. 17) represents an amount of work claimed from the 1176.5 B.T.U. of Intrinsic Energy. All of this area is not net, however, since the toll of the back pressure work in pushing the larger volume from the cylinder into the condenser must be exacted. The net work from the Intrinsic Energy is ABM. Its value cannot be computed directly, unless the law of the curve AB is known, and in this case it is not. But suppose it is assumed for the time being, as was done in Art. 23, that the quality of the exhaust steam, going to the condenser, is 93.5 per cent. Then if it is further assumed that no

heat is given to the steam or taken out of it while working in the engine, it is found that the heat content of the exhaust is 245.2 less than that of the steam supplied (Art. 23), and that amount has therefore gone into useful work. It is desired to find how much of the 245.2 B.T.U. is due to the external work effect, and how much is drawn from the intrinsic energy of the steam.

The area $E'EAA'$, Fig. 17, represents the whole of the 105.0 B.T.U. external work effect. However, only the area $FEAM$ is net work.

$$FEAM = \{(190 - .946) \times 144\} \times (3.002 - .01671)$$
$$= 81{,}270 \text{ ft.-lb. or } 104.5 \text{ B.T.U.}$$

The remainder, the difference between 245.2 and 104.5, which is 140.7 B.T.U. is represented by the area ABM, and is the amount of heat that is realized from the 1176.5 B.T.U. intrinsic energy.

38. How Heat Effects are Utilized in a Steam Turbine.—Fig. 18 is just the same as Fig. 15, except that the reciprocating engine of Fig. 15 is replaced by the impulse turbine of Fig. 18. Let the boiler and condenser maintain the pressure 190 pounds per sq. in. abs. and .946 pound per sq. in. abs., respectively, thus setting the same conditions as prevail in the power plant of Fig. 1. As a pound of steam is vaporized in the boiler and superheated in the superheater, it pushes the column of steam in the pipe in front of it, and the only escape for the steam in front is through the nozzle of the turbine. The external work effect appearing with the formation and superheating of the steam, would of itself, because of the continued urging from the rear, be manifested as velocity or kinetic energy of the jet at the mouth of the nozzle. But at the same time while the pressure in the nozzle is decreasing from 190 pounds to .945 pound, the steam has a further opportunity to expand, because of its own elasticity, or intrinsic energy. The result is a simultaneous utilization of the external work effect, and some of the heat of the stored or intrinsic energy.

Referring to Fig. 17, the entire area $FEABF$ is utilized, just as in the case of the reciprocating engine, but no distinct operation of the machine can be associated with the area $FEAM$, nor with the area ABM. The mass of steam appears at the mouth of the nozzle with a kinetic energy equivalent to the work represented by the entire area $FEABF$.

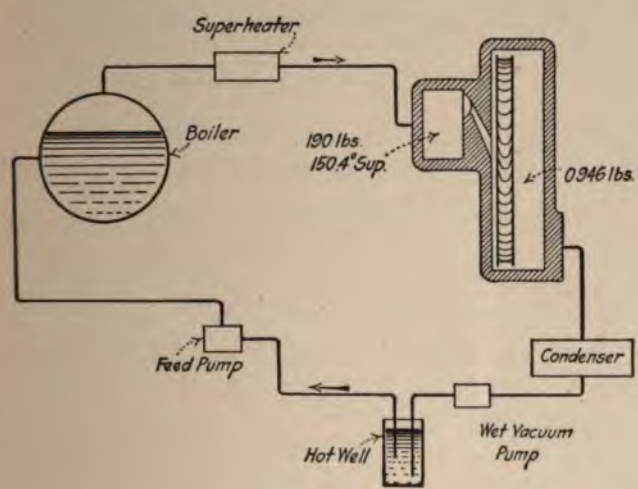

FIG. 18.

The work necessary to push the exhaust steam out of the turbine into the condenser is just the same as for the engine, viz., area $B'BFE'$; and the work of the feed pump in replacing the water in the boiler is, of course, the same in both cases. The operation as far as the working substance is concerned is cyclic, just as it was with the reciprocating engine. The turbine as a machine, however, operates upon the continuous flow, or non-cyclic principle.

Problems

1. Make up tables similar to the one shown in Art. 34, to show the distribution of the heat of a pound of steam, for the several cases specified below:

(*a*) Steam at 100 pounds abs. pressure and 100 per cent quality;
(*b*) Steam at 100 pounds abs. pressure and 95 per cent quality;

(*c*) Steam at 100 pounds abs. pressure and 100° superheat;
(*d*) Steam at 20 pounds abs. pressure and 98 per cent quality;
(*e*) Steam at 200 pounds abs. pressure and 99 per cent quality.

2. In a steam-heat engine (power plant), steam is furnished to the U
Utiliser at 165 pounds abs. pressure and 100° superheat, and is rejecte
the condenser at 2 pounds abs. pressure. The quality of the exhaust ste
assumed to be 94 per cent. How many heat units has each pound of st
contributed to useful work? Of the total number of heat units going
useful work, calculate the number derived from the "external work"
of the total heat, and from the "intrinsic energy" part, respectively.

CHAPTER VI

WHAT MAXIMUM PORTION OF THE HEAT SUPPLIED COULD AN IDEAL HEAT UTILIZER CONVERT INTO WORK ENERGY?

39. Actual Performance of a Turbine or Engine.—In preceding demonstrations, the actual performance of the turbine of the steam-power plant of Fig. 1 has been touched upon. In Art. 23 it was shown that the number of heat units realized in useful effect from each pound of steam was 245.2 B.T.U. and that the thermal efficiency of the plant was 20.0 per cent. But in arriving at these figures, a quality of exhaust steam of 93.5 per cent was assumed (Arts. 14, 19, 22, 23, and 37), whereas it has been pointed out that there is no simple and accurate instrument by means of which the quality at this point can be directly observed. The actual number of heat units converted into work by the turbine, and therefore the thermal efficiency, can be determined only by securing more observations than those that have already been indicated.

Suppose the turbine of Fig. 1 to be tested by measuring the power output of the turbo-generator, at the switchboard, and by determining the rate at which it consumes steam. The most direct and accurate means of effecting the latter measurement would be to rearrange the piping so that the wet vacuum pump would discharge the condensed steam into weighing tanks, in place of directly into the hot well. If a jet condenser happened to be employed in place of the surface condenser, a different plan would have to be resorted to, which will not be discussed here.

As a result of the additional observations, let it be assumed that it is found that the turbine is consuming

45,810 pounds of steam per hour, when it is delivering 3112 k.w. from the generator. The steam consumption, or water rate, as it is frequently called, per kilowatt per hour is therefore 14.72 pounds. It requires 3413 B.T.U. to maintain 1 k.w. for an hour. Each pound of steam therefore contributes 231.7 B.T.U. Assuming the generator efficiency to be 96 per cent, and the frictional loss in turbine bearings and stuffing boxes to be $1\frac{1}{2}$ per cent, each pound of steam contributes to the turbine shaft 245.2 B.T.U., which agrees with the results on the assumption of 93.5 per cent quality in the exhaust, because the latter figure was assumed in agreement with the test figures given above. The thermal efficiency of the plant is therefore just as worked out in Art. 23, viz., 20.0 per cent.

40. What Efficiency Could an Ideal Engine or Turbine Develop?—In the foregoing discussion, the performance of an actual turbine or heat utilizer has been considered. General experience, however, shows that no actual piece of mechanism can ever reach perfection. It is now natural and inevitable that there should come up the question, What could an ideal turbine or engine do? Could it convert the entire supply of heat into work, thus realizing 100 per cent thermal efficiency. If not, why not, and what portion of the heat supplied could it transmute into mechanical work?

It will be remembered that of the entire amount of heat supplied to the working substance, that division alluded to as the external work portion is nearly all easily realized in useful effect in the turbine or engine. (Art. 37.) Of that other general division of the heat supplied, the intrinsic energy, only a relatively small portion is recoverable as useful work. What portion is realized depends upon how the *expansion* occurs—how the pressure and temperature are let down from the upper limit to the lower.

41. Constant Entropy Expansion.—In general, the best way for steam to expand in the heat utilizer is at constant entropy, along the line FL in Fig. 19. During the opera-

tion FL no heat is added to the working substance and none is rejected from it. Such an expansion is called an adiabatic. Strictly speaking, the term adiabatic may be applied to a whole family of changes of which the constant entropy change is one. But almost invariably when the term adiabatic is used without qualification, the constant entropy change is the one that is meant.

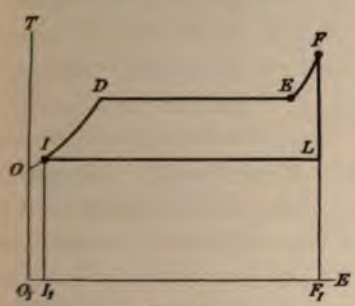

FIG. 19.

42. The Rankine Cycle.—In the steam-heat engine, as represented by Fig. 1, the turbine or heat utilizer performs but one operation in the cycle of the working substance, viz., the expansion of the steam. The line FL, Fig. 19, being now designated as the path of the expansion in the ideal steam-heat engine, the entire cycle of the working substance is completed, and is represented by $IDEFLI$. This is called the Rankine Cycle. The efficiency of the Rankine Cycle is the standard with which the performance of actual steam-heat engines may be compared to ascertain their excellences or their defects.

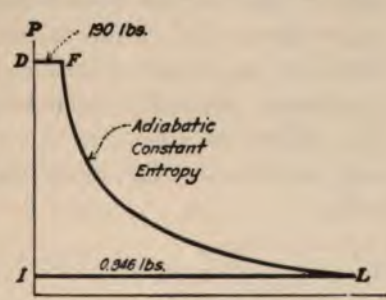

FIG. 20.

43. The Rankine Cycle on the *PV* Diagram.—The Rankine cycle is more familiar when represented on the PV diagram as in Fig. 20. The reader is to be reminded again that this picture is not an indicator diagram, but is a representation of the entire cycle of operations of the working substance. DF represents growth in volume resulting from evaporation of water and superheating of steam, all at constant pressure; FL is adiabatic expansion complete to

the back pressure; LI is condensation of the exhaust steam and ID is the elevation to boiler pressure in the feed pump. Water volumes are ignored in the diagram.

44. The Rankine Cycle on the Heat-entropy Diagram. The Rankine Cycle is pictured on the Heat-entropy diagram in Fig. 21. ID is the heating of water; DE is evaporation; EF is superheating at constant pressure; FL is adiabatic expansion; and LI is condensation of the exhaust steam. The area $IDFLI$ has, of course, no significance in this diagram.

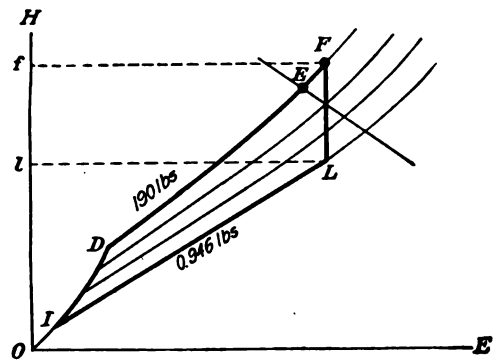

FIG. 21.

45. Heat Available for Work—Rankine Cycle.—Referring to Fig. 19, the heat supplied to a pound of the substance above 32°, when it arrives at the state F, is represented by the area O_1ODEFF_1; and the heat rejected to the condenser, measured above 32°, by the area O_1OILF_1. The difference between the two, area $IDEFLI$, is the heat which, in the ideal engine, goes into mechanical work.

Let H_1 = heat supplied above 32° = area O_1ODEFF_1,

H_2 = heat rejected in exhaust above 32° = O_1OILF_1.

Then $H_1 - H_2$ = heat available for useful work.

For the conditions of Fig. 1, viz., 190 pounds pressure and 150.4° superheat, H_1 is readily found to be 1281.5 B.T.U. The value of H_2 must be computed.

$$H_2 = h_2 + q_2 L_2$$

Where h_2 is heat of the liquid at exhaust pressure (.946 pound),

q_2 is quality of exhaust steam,

and

L_2 is latent heat of steam at exhaust pressure; h_2 and L_2 are readily found from the Steam Tables.

$$h_2 = 68.0$$

$$L_2 = 1035.6$$

q_2 must be found in a different manner from that which was employed in determining the quality of exhaust steam in the case of the actual turbine of the power plant. The initial condition of the steam is given and the nature of the expansion—the equation of the curve is specified. From these conditions, together with a knowledge of the exhaust pressure, the final condition of the steam as to quality and other properties can be established.

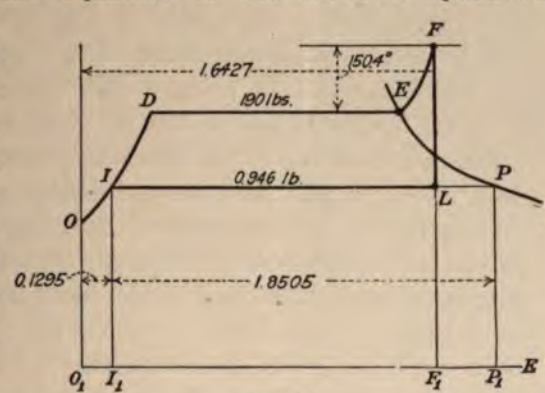

FIG. 22.

Referring to Fig. 22, point L represents the state of the steam at exhaust. It is necessary to determine the quality, q_2 at L. This can be done with the help of the steam tables and the figure.

The total entropy at F is 1.6427. The entropy at I—the entropy of water at 100°— is found to be .1295. The entropy of complete vaporization at 100° is $1.8505 = IP$. The change of entropy in vaporizing or condensing the fraction of a pound represented by q_2 at L is the distance IL. Now $IL = 1.6427 - .1295 = 1.5132$.

Therefore,

$$q_2 = IL \div IP = 1.5132 \div 1.8505 = .818 = 81.8 \text{ per cent},$$

$$H_2 = h_2 + q_2 L_2 = 68.0 + (.818 \times 1035.6) = 915.1.$$

$$H_1 = 1281.5$$
$$H_2 = \underline{915.1}$$
$$366.4 \text{ B.T.U. available for work.}$$

46. Thermal Efficiency of Rankine Cycle.

$$\text{Thermal efficiency} = \frac{\text{heat available for work}}{\text{heat supplied}}.$$

The numerator of the fraction has been determined in the preceding article and found to be 366.4 B.T.U. The next question is, what is to be the denominator of the fraction? Is it to be 1281.5 B.T.U.—the total heat in the steam above 32°? Reference to the plant of Fig. 1 will convince the inquirer that the heat source did not have to supply as much heat as 1281.5 B.T.U. because the water was initially at 90° when it first began to receive heat in the circuit. It would seem that the rather obvious answer to the question asked above would be that the denominator of the fraction should be the quantity of heat supplied above 90°. It must be remembered, however, that the ideal cycle is being considered here, and in the ideal heat engine, which would operate upon such a cycle, the condensed steam would not be cooled in the condenser, nor outside of it, to a temperature lower than the temperature of the exhaust steam. The function of the condenser would be to condense the steam, not to cool the condensate. The condensate should be returned to the heat source at the temperature of condensation, or 100° in the case of the plant used in illustration. The " heat supplied " or the " heat charged " is the heat content at 190 pounds pressure and 150.4° superheat, measured above 100°.

$$\text{Heat charged} = H_1 - h_2 = 1281.5 - 68.0 = 1213.5.$$

$$\text{Thermal eff.} = \frac{366.4}{1213.5} = .302 = 30.2 \text{ per cent.}$$

47. Rejected Heat.—Heat rejected, measured above 100° = $H_2 - h_2 = q_2 L_2 = 847.1$ B.T.U. In the Rankine Cycle the heat rejected consists only of latent heat. The heat of liquid is an asset as the working substance makes contact with the heat source again. The heat of vaporization in the exhaust steam is of no further value. At exhaust, the working substance is at a low pressure and large volume. In order to become useful in the heat utilizer again, it must be concentrated in volume and increased in pressure. This might be done mechanically, by a compressor for example; but the work expended would be as much as the work done. Nothing would be gained by such a course. One great advantage of the vapor heat engine lies in the fact that the enormous volume of the substance at exhaust can be reduced to almost zero by the simple process of condensation, and the highly concentrated substance, as liquid, can then be raised to the higher pressure level with but little work. But in concentrating the charge, the heat of vaporization in the exhaust steam—heat that was supplied from the source—must be thrown away.

48. Use of the Heat-entropy Chart.—The Heat-entropy chart, or Mollier diagram, is extremely useful in the solution of problems involving the determination of various heat quantities. Referring to the Heat-entropy chart found in the back of the Steam Tables, find first a point at the intersection of the 190-pound line, and the 150.4° superheat line. If any two of the properties of steam, represented on this chart, are known, a point can be located, and such other properties as the chart shows can be read off at once. Having located the intersection which corresponds to point F, Fig. 21, the heat content H_1 above 32°, can be read at f, on the scale at the left. For the exhaust condition, L, the pressure is .946 pound and the entropy is, by specification, the same as at F. Consequently two properties for the final condition are known, viz., pressure and entropy, and the point L can be located at the intersecting

lines. The heat content of the exhaust steam (above 32°) is then read at l, on the scale at the left.

49. Heat Content, External Work, and Intrinsic Energy. —It is convenient to refer to the steam as it enters the turbine as having so much *heat content.* Some care must be exercised here in order to avoid a misconception. The heat content of the steam as it is about to enter the turbine is not heat actually possessed within the steam. In fact, as has already been seen, heat content alludes to heat *supplied* to the working substance. It is during the process of the formation of the pound of steam in evaporation, and its further enlargement in volume during superheating, that the external work part of the heat passes directly into work by pushing other steam out of the way. The " old " steam, so to speak, serves as a connection between the applied force due to enlarging volume in the boiler and superheater, and the resistance offered by the piston in the engine cylinder, or the nozzles of the turbine, when in communication with the boiler, just as does a connecting rod between cross-head and crank pin; when the imaginary pound of steam arrives at the turbine, it has only intrinsic heat to give up for work. But simultaneously it is pushed along by a " younger " pound of steam just forming, so that the entire work done by the " old " pound of steam is a result of both the external work part and the intrinsic energy part of the total heat, although one effect is produced while the steam is in the boiler and superheater, and the other after the steam is locked up in the engine or turbine.

50. The Rankine Cycle Is Not the Most Efficient Cycle. —Although the Rankine Cycle has been alluded to as a representation of the performance of the ideal steam-heat engine, it is not the most efficient cycle possible for a heat engine. The efficiency of the Carnot Cycle, which is discussed later, or any other *reversible* cycle, is higher than that of the Rankine Cycle, and represents the maximum possible attainment for any heat engine.

For the steam-heat engine, other cycles than the Rankine may be imagined, that will result in higher efficiency. Referring to Fig. 23, suppose that, in place of expanding the steam at constant entropy from F to L, heat be added to the working substance while it is expanding, such as would be the case in a steam-jacketed engine cylinder. The operation is represented by FK. For the Rankine Cycle:

$$\text{Eff.} = IDEFLI \div I_1IDEFF_1.$$

For the supposed cycle:

$$\text{Eff.} = IDEFKI \div I_1IDEFKK_1.$$

whether or not the efficiency of the supposed cycle is greater than that of the Rankine will depend upon how much of the area F_1FKK_1, representing the heat added during expansion, is located above the exhaust line IP, and how much

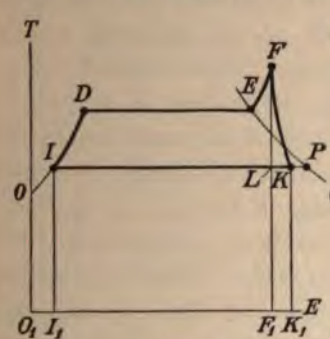

FIG. 23.

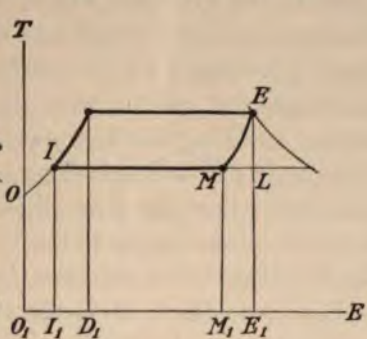

FIG. 24.

below. The area LFK is the part of the addition that is available for work, and F_1LKK_1 is the unavailable part. If LFK increases $IDEFLI$ more in per cent than F_1LKK_1 increases I_1ILF_1, then the efficiency of the supposed cycle is greater than that of the Rankine Cycle. The higher the superheat at F, that is, the taller the figure F_1FKK_1, the more likely it is that the efficiency of the supposed cycle will be higher than that of the Rankine Cycle.

If such an addition of heat be supposed for a case where

the steam is not superheated, it will be obvious at once, if the figure is drawn to represent the case, that the efficiency of a cycle with heat added during expansion is less than that of the Rankine Cycle.

Referring to Fig. 24, let heat be withdrawn during expansion, the operation being represented by EM. The area E_1EMM_1 is the total amount of heat withdrawn. If this heat be lost entirely from the circuit, then it is obvious that the efficiency of the cycle is less than that of the Rankine Cycle, because the area representing heat available for work, $IDEMI$, is less than for the Rankine Cycle $IDELI$, while heat that must be supplied—the denominator of the efficiency fraction—is just the same as for the Rankine Cycle.

Suppose, however, that the heat E_1EMM_1, Fig. 24, were returned to the working substance. The practical conception of this idea would be to return the condensate, delivered from the condenser of Fig. 1, through a jacket around the barrel of the turbine. Theoretically, enough heat might be taken from the expanding steam, in this manner, to bring the temperature of the condensate (feed water) back just to the boiling point. (It should be remembered here that the case illustrated in Fig. 24 is that of saturated steam supply to the turbine.) The area E_1EMM_1 is then just equal to the area I_1IDD_1. The only new heat supplied from the source, the boiler, is that for vaporizing the steam, viz., D_1DEE_1. The efficiency of such a cycle will be found, if worked out, to be superior to that of the Rankine Cycle. It is, in fact, equal to that of the Carnot Cycle.

The above examples are offered not so much to demonstrate the fact that the Rankine Cycle may be exceeded in efficiency by other possible steam cycles, as to emphasize the idea that the Rankine Cycle—the yardstick whereby the performance of steam-heat engines is measured—is, after all, somewhat arbitrarily chosen.

Problems

1. Calculate the thermal efficiency of the following cycle, making sketch of temperature-entropy diagram to illustrate operations: Steam furnished to heat utilizer at 175 pounds abs. pressure, 250° superheat; 60 heat units per pound of steam are added during expansion, by a jacket. The expansion is assumed to be along a straight line, FK, Fig. 23. Steam exhausts at .85 pound abs. pressure.

Compare the thermal efficiency of the above cycle with that of the Rankine Cycle.

2. Referring to Fig. 24, let steam be supplied at 190 pounds abs. pressure, 100% quality and exhausted at .946 pound abs. pressure. Let heat be transferred by a jacketing arrangement from the steam to the returned condensate, during expansion, so that the temperature of the condensate is raised to the boiling point corresponding to 190 pounds pressure. The expansion of the steam is represented by EM, and the area E_1EMM_1 will equal the area I_1IDD_1. Calculate the thermal efficiency of the cycle.

How does the efficiency compare with that of the Rankine Cycle for the same conditions?

How does it compare with the value of the ratio $(T_1 - T_2) \div T_1$, where T_1 is the absolute temperature of the steam at admission, and T_2 is the absolute temperature of the steam at exhaust?

CHAPTER VII

AVAILABLE, UNAVAILABLE, UTILIZED AND WASTE-ENERGY-LOSSES

51. Available and Unavailable Energy.—Of all the heat supplied to a pound of steam, it is now evident that it is possible to get only a portion of it into mechanical work. The remainder of the heat must be thrown away. These statements are true, no matter what kind of a heat utilizer may be employed, not even excepting a perfect engine. In the case of steam, an engine operating on the Rankine Cycle would be considered a perfect engine. It is not to be understood that the Rankine Cycle is necessarily the cycle of highest possible efficiency for any heat engine, or even for a steam engine. But the Rankine Cycle is predicated upon conditions which more nearly coincide with those under which a steam motor must operate, and for that reason is universally used as the standard in gaging the energy-transforming capacity of a steam utilizer.

In Fig. 25, FL represents an adiabatic expansion from an initial condition F. The area I_1IDEFF_1 is considered to be the heat quantity supplied. Of the entire amount, the portion $IDEFLI$ is converted into work by an engine operating on the Rankine Cycle, that is, by an ideal engine. Hence, the heat represented by the area $IDEFLI$ may be called the *available* portion of the heat, or more usually, the *available energy*.

The heat represented by the portion of the area below the line IL, that is the area I_1ILF_1 can not be touched even by a perfect heat engine. Consequently it may be called the *unavailable* portion of the entire heat supplied, or the *unavailable energy*.

Summarizing, it may be stated that of all of a given quantity of heat supplied to a working substance, only a portion can be converted into mechanical work; the remainder must continue as heat. These two divisions are named the available and-unavailable energies, respectively.

52. What Determines the Amount of Available and Unavailable Energy?—It is obvious that the heat utilizer (or turbine in the case of Fig. 1) has nothing to say about the amount of available energy in the steam furnished it. This is set by conditions wholly external to the turbine. The amount of available energy, in Fig. 25, might be increased by starting from a higher condition than point *F*, which would be a situation to be met by boiler and superheater. Or, the proportion of available energy can be made larger by lowering the boundary line *IL*, which is a matter depending partly upon the excellence of the condenser equipment, but ultimately upon the temperature of the cooling agent (water) employed in the condenser. The temperature of the cooling water depends, in turn, upon natural factors, such as geographical location and climatic conditions. The atmosphere is the great temperature leveler, tending to bring all things of a given locality to a uniform temperature.

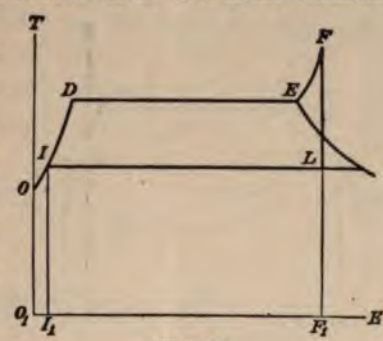

FIG. 25.

53. Utilized and Waste Energy.—The actual heat engine is necessarily less efficient than the ideal engine, or Rankine Cycle engine in the case of steam. The actual engine may be said to have a chance at the available portion of the total heat energy supplied. It converts what fraction of the available energy it can into useful work at its shaft. The available portion of the heat supplied is there-

fore to be thought of as being split into two divisions—one division disposed of in mechanical work at the shaft of the prime mover, and the other lost. The former may be termed the utilized energy; the latter, the waste energy. Waste energy is heat which was available, but through the imperfections of the utilizer, has been rendered unavailable.

54. Ultimate Disposition of Waste Energy.—In the case of the turbine of the Power Plant of Fig. 1, it has been shown that of the heat supplied to each pound of steam, 245.2 B.T.U were actually converted into work at the shaft of the turbine (Arts. 23 and 39). If the steam were used in a perfect steam turbine, it would expand adiabatically from 190 pounds pressure and 528°, to .946 pound pressure. From the Mollier diagram or the Steam Tables, it is found that 366.4 B.T.U. would be converted into work (Art. 45). 366.4 B.T.U. is therefore the Available Heat. The Heat Content of the supply steam is 1281.5, or the area O_1ODEFF_1, Fig. 26. But the heat charged to the turbine is the area I_1IDEFF_1 or

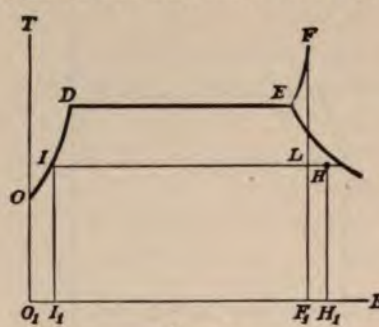

FIG. 26.

$$H_1 - h_2 = 1281.5 - 68 = 1213.5$$

The heat content of the condensed exhaust steam, h_2, is not charged to the engine because it can be reclaimed. It is represented by the area O_1OII_1.

$IDEFLI$, the Available Energy = 366.4 B.T.U.

F_1LII_1, the Unavailable Energy = 1213.5 − 366.4
= 847.1 B.T.U.

But out of the 366.4 B.T.U. available energy, only 245.2 B.T.U. actually appear as useful work. What has become of the remaining 121.2 B.T.U. which constitute

the Waste Energy? In answering this question, reference is made to Fig. 27. 1213.5 B.T.U. are supplied to the turbine in each pound of steam. By the circumstances of initial and final conditions, determined independently of the turbine, the 1213.5 B.T.U. is divided into available and unavailable portions as previously determined. Now all the energy that flows into the turbine must appear emerging from it in some form. If " radiation," which is a relatively small amount (usually not more than 1 per cent)

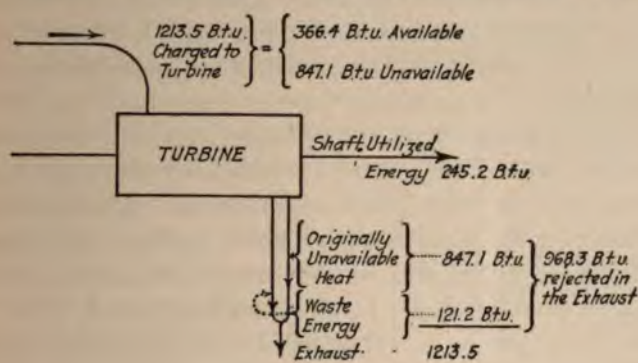

FIG. 27.

be excluded, then the outflow of energy can be sought in but two places, viz., at the shaft of the turbine, where it is in the mechanical or desired form, and in the exhaust steam. By observation or test, the energy emerging by way of the shaft is found to be 245.2 B.T.U. per pound of steam. Hence all the remainder of the 1213.5 B.T.U. or 968.3 B.T.U. per pound of steam is rejected to the cooling water of the condenser. The heat of the exhaust comprises the 847.1 B.T.U. of originally unavailable heat and the 121.2 B.T.U. of waste energy, which thus appears augmenting the unavailable portion of the heat supplied. The turbine, by its imperfection, has dissipated 121.2 B.T.U. of the available heat entrusted to it, thereby adding to the unavailable heat.

In Fig. 26, the area $F_1 L H H_1$ must be added to $I_1 I L F_1$

to form the area H_1HII_1, which represents the heat rejected in the exhaust. The strip H_1HLF_1 represents the waste energy; it came originally from the area $IDEFLI$. The utilized portion of the available energy cannot be be represented on the diagram.

55. How May Heat Energy Pass from the Available to the Unavailable State?—How does a portion of the available heat pass into the unavailable state? Or to be explicit, how did the 121.2 B.T.U. of the preceding paragraph become unavailable. The actual turbine appears to conform to one requirement of a Rankine Cycle, viz., that no heat is added or rejected during the process of expansion. "Radiation" losses have been neglected; hence no heat, as heat, is removed while the steam is in the turbine. Yet no one is likely to be misled into thinking that because the operation is "adiabatic" in the sense of non-transference of heat, the turbine is going to perform perfectly, and convert the entire amount, 366.4 B.T.U., of the available heat into useful work. On the other hand, it is not clear at once just why it should not do so. The best explanation is to be found in the throttling calorimeter.

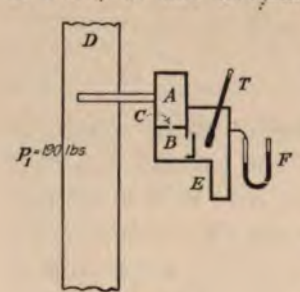

FIG. 28.

56. The Throttling Calorimeter.—The throttling calorimeter, Fig. 28, consists of a high- and a low-pressure chamber, A and B, respectively. A is in communication with the steam in pipe D, at the pressure P_1. The steam is exhausted from B into the atmosphere through the pipe E. Steam passes from A to B through the orifice C. The pressure P_2 in B is measured by the manometer F, and is but slightly more than atmospheric. The thermometer T indicates the temperature in B. The entire instrument is usually well lagged so that loss of heat to the atmosphere is negligible.

Let this calorimeter be supposed to be located upon the steam line between the boiler and superheater of Fig. 1 with a view to ascertaining the quality or moisture in the steam as it is made by the boiler. The pressure in the pipe is 190 pounds abs., as shown by a gage attached near by, together with the barometer reading, which is found to be 29.53 in. The thermometer indicates 298° F., and the manometer reads 2.8 in. of mercury. The pressure P_2 in the chamber B is therefore 15.9 pounds per sq. in. abs. If the steam in B were saturated, its temperature would be 216°. There is actually therefore a superheat of 82° and the heat content of the steam in B is found to be 1189 B.T.U.

If the calorimeter be examined in the same critical manner in which the turbine of Art. 54 was observed, it will be found that there is an inflow of energy whose value is not yet known. It is to be expected that all the energy that goes in will be accounted for in the sum of the energy quantities that emerge from the instrument. But examination reveals the fact that there is no shaft or other channel for the escape of energy in the form of work; and since atmospheric loss is neglected, the only channel for the outflow of energy is the exhaust pipe, where it is all in the form of heat. Hence the fundamental conception of the throttling calorimeter, viz., that the heat content of the entering steam is equal to the heat content of the emerging steam.

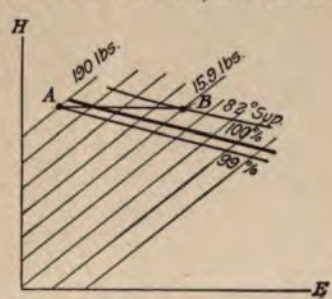

FIG. 29.

If the entering steam contains moisture, then its heat content is expressed by the equation:

$$H_1 = h_1 + q_1 L_1 = H_2$$

H_2 has been calculated above from observation and is found to be 1189 B.T.U. As a knowledge of P_1 enables the

determination of L_1 and h_1, from the steam tables, q_1 remains the only unknown factor, and is easily computed and found to be .99 or 99 per cent.

The solution of the problem is easily accomplished by means of the heat chart, Fig. 29. The point B is located at the intersection of the 15.9 pound pressure, and the 82° superheat lines (interpolating). The heat content at B is the same as at some point A on the 190 pound pressure line, found by the intersection of the horizontal line through B with the 190 pound line.

The point of intersection, A, represents the condition of the steam in the pipe at high pressure, and is found to lie upon the 99 per cent quality line.

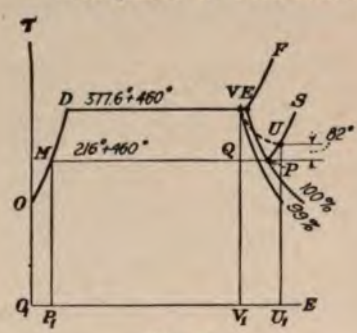

FIG. 30.

57. Throttling Destroys Availability of Energy.—In Fig. 30, point V represents the initial state of the steam in the pipe. The heat content of the steam is O_1ODVV_1. The available portion of the heat is $MDVQ$. After passing through the calorimeter, its condition is represented by U on the superheat section of the constant pressure line $OMPS$ (15.9 pounds) at 82° superheat. The path of change may be thought of as VU, although such a line must not be confused with a line of change that would result from applying heat during the operation. In the latter case, the area V_1VUU_1 would represent the heat supplied. In the case of throttling, the area V_1VUU_1 has no significance.

The point of importance is that the heat content at U is the same as at V.

$$O_1OMPUU_1 = O_1ODVV_1$$

None of the available heat $MDVQ$ has gone into useful work. After the operation, all of the heat is found in the

exhaust at the lower pressure. The originally unavailable heat has been augmented by *all* the originally available heat.

$$V_1QPUU_1 = MDVQ$$

Throttling is the letting down of pressure, and growth of volume, without any net useful work being done. It destroys no absolute quantity of energy, but it destroys availability. Whenever it occurs in any of the operations of a heat engine, there results a waste of available heat, with a consequent reduction of efficiency.

58. Throttling, Steam Friction, Wire Drawing, Pressure Reducing.—The throttling process is known by various names. Frequently the principle is employed to regulate the amount of steam or working fluid used by an engine. Throttling in this case is deliberate or intentional. Some advantage of construction or operation is sought, that is expected to counterbalance the loss of available heat occasioned by throttling.

When steam or any other fluid flows through a pipe, it is subjected to a loss of pressure. The cause of the pressure loss is referred to as fluid friction; but thermodynamically, it is not different from what takes place in the throttling calorimeter. The heat content of the steam emerging from a section of pipe is just the same (excepting for radiation) as it was when it entered; but the pressure is less.

When steam squeezes through ports or openings that are too small to accommodate the amount, it does so with a loss of pressure, and the operation is called *wire-drawing;* in reality, it is throtting.

When high-pressure steam is passed through a reducing-pressure valve to lower its pressure to be suitable for steam-heating purposes, the process is referred to as " reducing the pressure." This again is an illustration of a throttling operation.

All throttling processes are wasteful. Availability of energy is destroyed—availability that can never be restored. A popular misconception is that, in the case of steam,

throttling will superheat the steam, and if there is superheat, there must be, somehow, more heat or more availability than there was before. But it must be remembered, in the first place, that there is no more heat after the operation than before; and in the second place, throttling may not even result in superheating. Let a problem be solved assuming steam at 190 pounds pressure and 94 per cent quality, and throttling to 15 pounds abs. pressure; it will be found that the steam is still wet after throttling. In the third place, although the heat content after throttling is the same as before, and there is a high degree of superheat, can any work be done with the steam in its final state when there is no pressure head to stir it into action?

59. How Throttling Destroys Availability of Energy.—Take the case of the throttling calorimeter again, of which Fig. 31 represents the elements. In passing through the orifice C some of the available energy changes to kinetic energy (which is a form of mechanical energy), appearing in the velocity of the jet emerging into B. It may even be assumed as an extreme case that *all* the available energy is converted into kinetic energy, in which case the orifice would be a perfect nozzle. The steam has therefore been used in a manner 100 per cent efficient, since all its available energy is converted into mechanical effect.

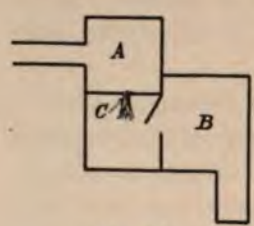

FIG. 31.

If a turbine wheel were placed in the path of the jet, and the wheel were 100 per cent efficient, then all the kinetic energy would appear as useful effect in the shaft of the wheel and the whole apparatus would be a perfect steam turbine.

In the case of the calorimeter, there is no wheel present to claim the velocity energy of the jet. As the outlet of chamber B is arranged, the jet cannot pass out directly, even if there were no baffles such as are shown. It is

obvious that the jet will be broken up, by the turns and interferences to which it is subjected, and the steam will emerge filling the discharge pipe, which is of such large section that the velocity is relatively low. The kinetic energy of the orifice jet has been destroyed, and in the destruction the mechanical form of energy has reverted to the heat form. This reconversion of work energy, which originally came from heat, back into the heat form is called *reheating*. The entire process of throttling may be conceived of as first the formation of velocity or kinetic energy from heat energy by the drop in pressure; and second, reheating, which is the reconversion of the kinetic energy into heat.

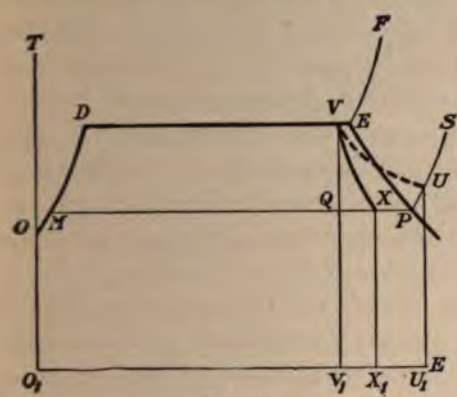

FIG. 32.

60. Representation of Throttling Process.—What takes place in a calorimeter is represented in Figs. 32 and 33. If the orifice were a perfect nozzle, there would be first adiabatic expansion along VQ, resulting in the formation of velocity, and then constant pressure reheating along QPU, as the velocity is destroyed and reconverted into heat.

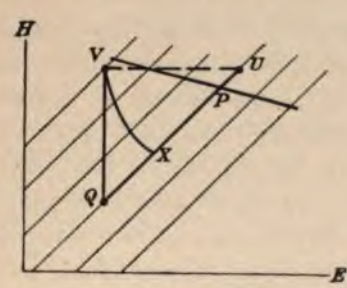

FIG. 33.

The actual process of throttling in a calorimeter, or any other apparatus, is not truly represented by the route VQU. In the calorimeter the steam passes through an orifice which is far from being a nozzle of 100 per cent

efficiency. By this is meant that not all the available energy of the steam will appear as velocity as the steam emerges from the orifice. In fact, the process of reheating goes on to some extent, within the orifice itself, simultaneously with the decline in pressure. The action is indicated by the path $VXPU$, Figs. 32 and 33. As the pressure starts to drop from V, some velocity is imparted. But as soon as velocity is created, frictional effects appear, which means the reconversion of some of the kinetic energy into heat. In the meantime, further drop in pressure is taking place; and simultaneously, further reheating. By the time the steam has emerged into chamber B of the calorimeter, Fig. 31, it has less velocity and higher heat content than it would have had, had the orifice been a perfect nozzle. Whatever kinetic energy the jet has as it emerges into B, is then converted back into heat. Expansion takes place along VX, accompanied by some friction losses and reheating. The area V_1QXX_1, Fig. 32, is the amount of heat restored to the steam by the frictional effect within the orifice. The reheating resulting from the destruction of velocity of the jet, is represented by XPU. The heat restored to the steam is the area X_1XPUU_1 (Fig. 32). The final state U is the same, no matter what intermediate relation exists between the reheating within the orifice and in the chamber B.

61. Losses in a Steam Turbine.—In a steam turbine nearly all the losses are of a throttling nature, in that they can be traced to frictional effects in which the steam is involved, and from which reheating results. The nozzles correspond to the orifice of the throttling calorimeter and although they are made as nearly perfect as is practicably possible, yet some friction is developed within them. In the blades, further frictional effects occur, and even after the steam is released from them, the revolving wheels and blades contribute still more to its accumulation of heat from frictional losses. The unavailable heat of the steam is progressively augmented by heat that comes from the

originally available portion of the supply, after having existed in the temporary state of mechanical energy.

62. Losses in a Steam Reciprocating Engine.—Steam friction loss is a necessary accompaniment of velocity or flow. The steam turbine is a velocity machine. Hence its peculiar susceptibility to this loss.

The steam reciprocating engine, on the other hand, is a direct-pressure machine. Its operating principle does not involve the necessity for the formation of velocity except for the incidental purpose of getting the steam into and out of the working chamber. Frictional effects accompany the creation of these incidental velocities, and the kinetic energy represented by them is quickly returned to the steam in the form of heat. This loss is called "wire drawing" and as has been seen, is in reality throttling.

But after all, wire drawing, while an appreciable loss, is not a serious one in most engines, and is in no degree comparable in magnitude to the total losses met with in a turbine. Is it therefore to be concluded that the reciprocating engine is inherently superior to the turbine? Experience does not bear out such a conclusion. The reciprocating steam engine is subjected to another very serious loss, which is peculiarly its own.

63. Initial Condensation and Re-evaporation.—Let Fig. 34 represent a steam-engine cylinder. With the piston at the extreme right, just after the steam in the head end has completed its expansion and has been released, the exhaust stroke begins and the remaining steam is forced out of the cylinder. The temperature of the steam is at its lowest and it is tending to cool the walls of the cylinder, the piston, ports, and valve. When the piston reaches the end of its

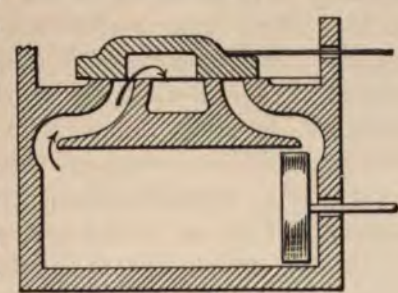

FIG. 34.

stroke, at the left, the valve opens allowing hot live steam to pour in upon the relatively cool surfaces, and partial condensation occurs. In the meantime, the piston moves out, and at the proper time cut-off takes place. The mixture of steam and water of course occupies the volume behind the piston. If the steam were dry or of the same quality as upon entering, its volume would be sufficient to push the piston a good deal farther than its actual position at cut-off. On the PV plane, Fig. 35, the steam may be thought of as originally occupying the volume MG, but on account of condensation its volume is actually only MK at cut-off. As expansion takes place, the pressure of the steam drops, and with it the temperature. At first, the temperature of the steam is higher than the average temperature of the walls, and condensation, due to abstraction of heat from the steam, continues, but at a lessening rate. But after a time the temperature of the steam falls below the average temperature of the confining walls the flow of heat reverses, the steam gains heat from the metal during the remainder of expansion to L, through release and exhaust to Q which results in the re-evaporation of some of the moisture. The cycle of heat interchange between steam and metal is referred to as *initial condensation and re-evaporation.*

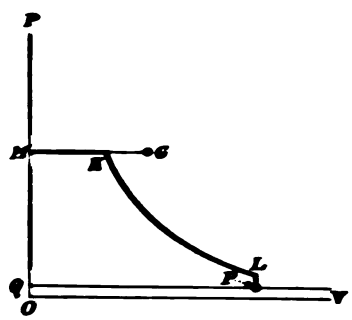

FIG. 35.

64. Why Initial Condensation and Re-evaporation Results in a Loss of Availability of Energy.—In one respect, the phenomenon just described, occurring in an engine cylinder, is like the frictional or throttling loss of a turbine. In both cases a certain amount of energy is diverted at or near the beginning of the process, and later this energy is returned in full, or nearly so, to the working medium as

heat. There is no net loss of energy quantity in either case.

The difference between initial condensation and throttling losses lies in the fact that in the former the energy is diverted as heat, remains stored as heat in the walls of the working container, and is finally returned to the working substance as heat. In the case of throttling, heat energy is first changed to kinetic energy; friction results, causing a transformation of work into heat energy, which is ultimately returned to the working medium.

Although there is no loss of total energy quantity in either case, there is loss of available energy in both. How initial condensation and re-evaporation results in a loss of available energy can be well pictured on the Temperature-entropy diagram.

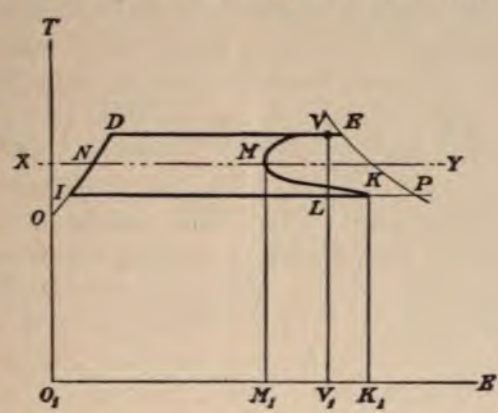

FIG. 36.

In Fig. 36 let V represent the state of the steam as it enters the engine. IP is the exhaust condition furnished by the condenser. Let XY represent the average temperature of the cylinder walls. When the steam is at a higher temperature than XY, heat will flow from steam to metal, and conversely, when the steam is at a temperature lower than XY, the direction of flow is reversed.

Starting at condition V, the abstraction of heat begins immediately upon the introduction of the steam into the cylinder. After cut-off, the steam begins to expand, lowering in pressure and temperature, and doing work upon the piston. The abstraction of heat by the walls continues

simultaneously, as long as the temperature of the steam remains above XY. The operation so far may be represented by VM. It is a case of expansion, accompanied by the withdrawal of heat. If no other losses have occurred in the cylinder, the work done by the steam on the piston is the area $NDVM$. The amount of heat abstracted from the steam and stored in the metal of the cylinder and piston is V_1VMM_1.

As the pressure of the steam, and therefore its temperature, continues to drop, from state M, the flow of heat will reverse, the metal giving out heat to the steam. The remainder of the expansion is represented by the line MK. The amount of heat restored to the steam is M_1MKK_1. Since the amount of heat given back to the steam by the metal during the latter part of the expansion, is just equal to that taken from the steam during the early part of the expansion (neglecting " radiation " loss), the area M_1MKK_1 = area V_1VMM_1. But this important point is to be observed—the area V_1VMM_1 is taller than the area M_1MKK_1. Hence, for equality of areas, the widths of the strips cannot be equal. The entropy change from M to K is greater than that from V to M. The final state K, after expansion, will be to the right of the initial state V, although no net amount of heat has been added. The quality of the exhaust steam is higher at K than it would have been after adiabatic expansion to L. Condensation and re-evaporation have operated to take away some of the area $IDVLI$, and add to the area representing the heat in the exhaust steam by the amount V_1LKK_1. The process has rendered some of the available energy unavailable. The final effect is exactly the same as that which follows throttling.

The steam turbine is practically free from initial condensation losses, because at any given point, the steam is always at one temperature, for a given load, and the metal in contact with it assumes that constant temperature. There is no periodic interchange of heat. On the other hand, this advantage on the part of the turbine is offset

by the fact that it must contend with large steam-frictional losses, because it is a velocity machine.

The Uniflow reciprocating engine is an attempt to approach the favorable condition of the turbine with respect to condensation and re-evaporation loss, by keeping the steam moving in one direction as far as may be possible with a reciprocating mechanism. The cool exhaust steam is not pushed out through the same passages, or even the same end of the cylinder as that at which the hot steam enters, but passes out of the cylinder, after its work is done, through ports uncovered by the piston itself. The idea of the uniflow engine is the most important thermodynamic achievement in the development of the steam engine since the introduction of compounding.

Problems

1. Determine the available and unavailable heat per pound of steam for each of the cases specified below:

(*a*) Initial pressure and quality, 175 pounds per sq. in. abs. and 99% respectively; exhaust pressure, 1 pound abs.

(*b*) Initial pressure and quality, 175 pounds per sq. in. abs. and 99% exhaust pressure, 16 pounds abs.

(*c*) Initial pressure and quality, 16 pounds per sq. in. abs. and 96%; exhaust pressure, 1 pound abs.

(*d*) Initial pressure and superheat, 175 pounds per sq. in. abs. and 200°; exhaust pressure, 1 pound abs.

Compare the answers of *a*, *b*, and *c*, and note the relation between pressure drop and available energy in the high- and low-pressure regions. Also compare the answers of *a* and *d*.

2. Steam is furnished to an engine at 140 pounds per sq. in abs. and 98.5% quality. The back pressure is 5 pounds above atmospheric, or 19.5 pounds abs. The engine uses 32.5 pounds of steam per indicated horse power per hour, as shown by a test.

(*a*) How much heat is available to the engine, and how much unavailable, per pound of steam?

(*b*) How many B.T.U. does each pound of steam contribute to useful work in the cylinder?

(*c*) What is the thermal efficiency of the engine, and what is its efficiency in terms of the available energy?

(*d*) Neglecting "radiation," how much waste energy is there and what is the quality and heat content of the exhaust steam?

3. Steam is furnished to a turbine at 150 pounds per sq. in. abs. and 99% quality. At a given load on the turbine, the steam is throttled by the governor to 90 pounds pressure, before it enters the turbine proper. The exhaust pressure is 2 pounds abs. Calculate the amount of available energy wasted by throttling. Use the Mollier Chart in the solution and illustrate the manner of its use by a sketch.

CHAPTER VIII

VAPOR REFRIGERATION

65. The Ammonia Compression Machine.—In Fig. 37, ammonia in the vapor state enters the compressor at *A*. Its pressure, in this particular case, is shown to be 25 pounds abs., and the temperature 30°. The compressor, which is mechanically exactly the same thing as an air compressor, delivers the ammonia at *B*, at 200 pounds pressure and 210°. The ammonia is conveyed through the pipe to *C*, where the temperature is 190°, the drop of 20° being due to loss of heat to the atmosphere. On its way, the vapor is passed through an oil separator to remove particles of oil or water that may be carried along from the cylinder.

The ammonia next traverses the condenser and emerges at *D* at 80° in the liquid state. Condensation is accomplished by the circulation of water at whatever may be the "natural" temperature of the locality and the season. The receiver is a reservoir for storage of the liquid which pauses here until it is called upon to make another circuit of the system. From the receiver the ammonia passes to the expansion valve *V*, through which the pressure is reduced. The valve *V* is nothing more than a throttling, or pressure reducing, valve. The pressure following the reducing valve is shown to be 25 pounds (abs.) by the gage near the compressor.

In the expansion coils, the liquid ammonia is evaporated, an operation which demands a supply of heat from some source. The source here is the brine. The brine thus cooled, returns to the ice cans and gets another load of heat to convey to the ammonia. This operation goes on indefinitely, the brine acting as a heat vehicle, loading up at one

point and discharging its load at another. In place of the ice-making equipment, the brine may serve to cool a room, by taking heat from the air; or the ammonia coils themselves may be placed directly within a room which is to be kept cool.

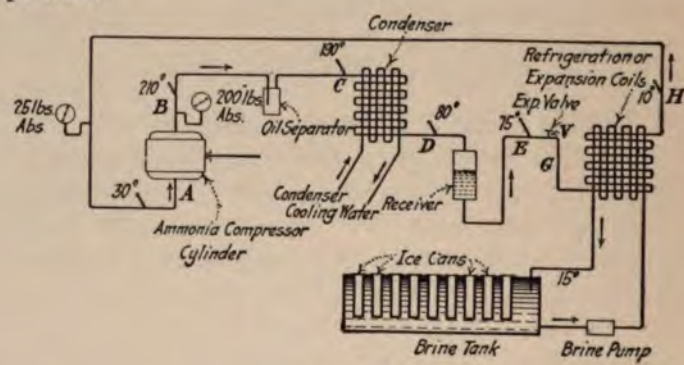

FIG. 37.

The ammonia finally leaves the expansion coils at H, completely evaporated, with a temperature as shown by the thermometer, of 10°. On its way back to the compressor the effect of the atmosphere is to warm the ammonia; the " radiation " effect is negative. The ammonia is returned to the compressor at A, with a temperature of 30°, thus making the complete circuit of the system.

66. The Properties of Ammonia.—Just as was the case in the study of the steam-power plant, an investigator is at once confronted with the necessity of a knowledge of the properties of the working substance. As used in the refrigerating system described, ammonia is a substance which is sometimes a liquid, sometimes a vapor, and sometimes a mixture of the two. In that respect it is identical with the working substance of the steam-power plant. It will therefore behave in a similar manner, and the properties can be worked out from experimental data, in a form parallel to that of the Steam Tables. Abbreviated tables of the properties of saturated and superheated ammonia may be found in the Marks " Mechanical Engineers' Hand-

book" (McGraw-Hill Co.). Properties of various other refrigerating agents are also given there. For complete tables of the properties of ammonia, the reader is referred to "The Properties of Steam and Ammonia" by Goodenough (Wiley & Sons).

With the pressures and temperatures indicated by the gages and thermometers, together with the ammonia tables, the properties of the working substance at the several points in the circuit can now be obtained.

67. Representation of Cycle on the Temperature-entropy Plane.—The changes in state of the ammonia can be very profitably pictured on the temperature-entropy plane (Fig. 38). Start with the ammonia at A, just as it is about to enter the compressor. The temperature is 30° F. and the pressure is 25 pounds abs. Referring to the Ammonia Tables, the temperature of saturation is found to be −7.2°. The ammonia is therefore superheated 37.2° at A and the entropy is 1.235 (Goodenough's Tables). The zero of entropy is arbitrarily chosen for the Tables at 32° as it was for steam.

At B the pressure is 200 pounds abs. and the temperature is 210°. The saturation temperature is found to be 95.9°, so that there is superheat here also, to the extent of 114.1°, and the entropy is 1.121. The entropy at B is seen to be less than at A. Is this consistent with the nature of the compression from A to B? It is, because the compressor cylinder is water-jacketed, which means that, although the energy content is increased, heat is actually withdrawn during the operation by the amount A_1ABB_1. Only heat quantities are pictured directly on the TE diagram, and A_1ABB_1 represents the heat carried away by the jacket water per pound of ammonia. From B the ammonia is lowered in temperature to C by loss of heat, B_1BCC_1, to the atmosphere and is then cooled and condensed, leaving the condenser at state D, as a liquid at 80° temperature, but still at 200 pounds pressure. Point D is on the liquid line at a temperature below the saturation tem-

perature; and the point E is 5° lower, on account of the atmospheric cooling. The next change is a reduction of pressure through the expansion valve V. This will be recognized as a purely throttling action. Hence the total heat is the same at G as at E.

$$h_1 + q_1 L_1 = h_2 + q_2 L_2; \text{ but } q_1 = 0$$

Hence

$$q_2 L_2 = h_1 - h_2 = O_1OEE_1 - (-M_1MOO_1) = M_1MEE_1$$

or

$$M_1MGG_1 = M_1MEE_1$$

From the ammonia tables,

$$h_1 = 47.8; \; h_2 = -41.3; \text{ and } L_2 = 577.8$$

Whence $q_2 = .154$ or 15.4 per cent.

The entropy at M, with respect to O, is, from the tables, $-.0876$. The entropy from M to G is $q_2 L_2 \div T_2 = .154 \times 577.8 \div 452.8 = .1986$. The coordinates of the point G are thus established.

From G to H, the ammonia receives heat from the brine. Evaporation takes place, and finally some superheating. A small amount of heat is further absorbed from the air, represented by the area H_1HAA_1. From A, the ammonia begins the cycle of operations over again.

68. Heat Quantities.—Referring to Fig. 38, the area A_1ABB_1 is the heat removed by the compressor cylinder-jacket water. B_1BCC_1 is the heat lost to the surrounding air. In this case, such loss of heat is a gain in economy since the next step is a continuation of the operation. What heat is lost to the air, does not have to be removed by the condenser. The condenser removes the entire amount of heat C_1CDD_1. More heat D_1DEE_1 is again advantageously lost to the air from the condensed ammonia. It has already been shown that there is no change of heat content in passing through the expansion valve. The ammonia receives from the brine the amount of heat G_1GHH_1, and it is the value of this area that expresses the

refrigerating capacity for each pound of ammonia circulated. The area H_1HAA_1 is another atmospheric effect, and although it is a gain in heat, it represents a loss in efficiency, since whatever heat is gained here, must be taken out elsewhere in the restoration process.

69. Refrigerating Capacity.—The capacity or output effect of a refrigerating plant is expressed in terms of B.T.U. per hour transferred from the brine to the ammonia; or "tons refrigeration per twenty-four hours." A "ton" refrigeration means the removal of a sufficient amount of heat by the ammonia to freeze a ton of water at 32° into ice at 32°. That is, a "ton" is equal to $2000 \times 144 = 288{,}000$ B.T.U.

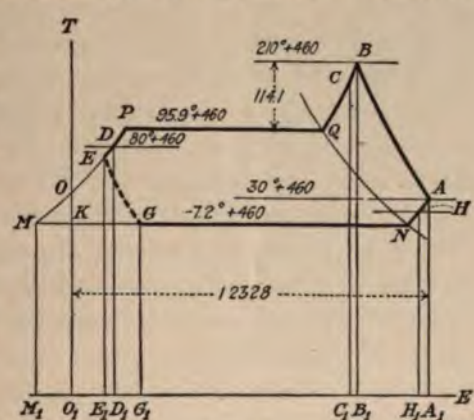

FIG. 38.

The capacity developed by a plant will depend upon two things, viz., the amount of heat removed per pound of ammonia (the area G_1GHH_1, Fig. 38); and the rate at which ammonia is handled.

Referring again to Fig. 38, the heat content at H is 546.5. The heat content at G is 47.8. Hence the area $G_1GHH_1 = 546.5 - 47.8 = 498.7$ B.T.U. In order to produce a ton of refrigerating effect (per twenty-four hours), it would be necessary for the plant to handle $288{,}000 \div (498.7 \times 24) = 24.1$ pounds of ammonia per hour.

70. The Refrigerating Coil is an Ammonia Boiler.—When allusion is made to that section of the ammonia refrigerating plant in which the brine is cooled by the evaporation of ammonia, as the *refrigerating coil*, the apparatus is viewed from the standpoint of the effect which it is

desired to produce. But viewed from the standpoint of the ammonia, the refrigerating coil is a boiler. Heat is absorbed by the liquid from a surrounding warmer substance, and the liquid is thereby converted into a vapor, which is exactly what happens in a steam-boiler plant. The brine corresponds to the hot gases. Both are cooled in the operation which takes place. Either apparatus is a *boiler* when the effect upon the confined volatile substance is considered; either is a *refrigerator* when the operation is viewed from the standpoint of the substance which furnishes the heat.

Liquid ammonia is fed into the ammonia boiler, just as water is fed into a steam boiler, but with this difference; the temperature of the ammonia liquid is higher than that of the vaporization pressure carried. It is as though water at a temperature of 400° F. were fed to a steam boiler carrying 100 pounds steam pressure. In place of heat having to be applied to raise the temperature of the feed to that of vaporization, the feed liquid possesses an excess amount of heat which it contributes toward vaporizing itself. It is a common error or at least an incomplete statement, to say that the liquid ammonia " flashes " into vapor on passing through the expansion valve. It is true that the excess heat of liquid is there applied to vaporizing some of the liquid itself. But the proportion of vapor by weight is the ratio $MG \div MN$ (Fig. 38), from which it is obvious that, of the entire weight, after passing the expansion valve, liquid constitutes by far the greater proportion.

71. Pressures in the System.—It is now pertinent to inquire what pressures it is necessary to carry in the two divisions of the system. The condensation temperature of the ammonia in the condenser must be higher than that of the cooling agent employed, which is water at " natural " temperature, by an amount sufficient to insure a fairly rapid flow of heat through the condensing surface. Consider cooling water at 70°. If the ammonia is to be condensed at the pressure of 200 pounds abs., as indicated on

the diagram of Fig. 37, the temperature of condensation will be 96°, which would allow a temperature drop of 26°. Any lower condensation pressure would result in a less temperature drop, and slower action would occur. In winter time, when the cooling water temperature is lower, the condensation pressure can be lowered correspondingly.

The other pressure, usually called the " suction " pressure, which exists in the refrigerating coils, is fixed by the rate of evaporation, which in turn depends upon the " load " or amount of cooling to be done. If the plant has been operating at a given load, and the load is then lessened the brine will not receive as heavy a supply of heat as the ammonia takes away from it in the coils. The immediate tendency is for the temperature of the brine to lower. This must be remedied, else the temperature of the brine will be gradually reduced until it may freeze and clog the coils, or even rupture them. The rate of evaporation, that is the number of pounds of ammonia handled per hour, must be reduced. Obviously, this can best be done by means of the expansion valve. The vapor within the low-pressure section of the circuit becomes more attenuated, with the result that the weight drawn into the compressor per stroke is less and less, until it finally becomes exactly equal to the amount passing through the expansion valve, when equilibrium is restored. The ideal method of control is to have the expansion valve automatically operated by a thermostat on the brine circuit. The suction pressure carried is an indication of the load on a plant. If the load is light, the pressure is low, and vice versa.

72. Other Working Substances for Refrigerating Machines.—What are the properties that make ammonia so widely used as a refrigerant? In the first place, its properties happen to be such that its boiling-point, for most refrigerating temperatures, is accompanied by a very moderate pressure; and again the higher pressure that must be employed to bring the condensation temperature well

above that of ordinary cooling water is also very moderate.

Steam and water may be used as are frigerating agent. Steam can be made at 32° by creating a low enough pressure. In fact, water vapor can be generated at temperatures below 32° directly from the ice—a phenomenon called sublimation. The evaporation of brine will enable the attaining of temperatures below 32°, provided a sufficiently low pressure can be established and maintained. From the Steam Tables, the pressure corresponding to 32° is .0886 pound per sq. in., or 29.82 in. vacuum. Steam refrigerating plants have been built and are in successful operation.

Other agents that are used to a considerable extent are carbon dioxide (CO_2) and sulphur dioxide (SO_2) . Carbon dioxide has the advantageous property of being non-poisonous, and therefore not dangerous if it should accidentally escape from the system. For this reason it is much used for refrigerating purposes on shipboard and in office buildings. On the other hand, an extremely high compression pressure is necessary to bring the condensation temperature to a point where liquefaction may be effected by water at natural temperatures. A pressure of over 1000 pounds per sq. in. is required to raise the condensation temperature to 85° F. The pressures required by sulphur dioxide for refrigeration purposes and liquefaction of the gas are even more readily attainable than those necessary for ammonia, a fact that has made SO_2 a very popular agent for use in small or domestic refrigerating machines.

The air refrigerating machine operates upon a wholly different principle from that of the vapor machine, and will be taken up later (Art. 115).

73. The Ammonia Absorption Machine.—Referring to the description of the ammonia compression plant, Art. 65, and Fig. 37, it is apparent that the real effect of the whole system is accomplished in the refrigerating coils. All the rest of the apparatus is auxiliary to this, and may be considered as merely participating in the restoration of the

ammonia to the liquid state, at low pressure, so that it can perform its function over again in the refrigerating coils.

In the Absorption Machine, compression is effected almost wholly by thermal processes, by taking advantage of the remarkable capacity of water to absorb ammonia at moderate temperatures, and the rapid reduction of this capacity with increase of temperature. Fig. 39 illustrates the absorption plant in a general way. Heat, applied in the generator by the steam coil, drives ammonia vapor out

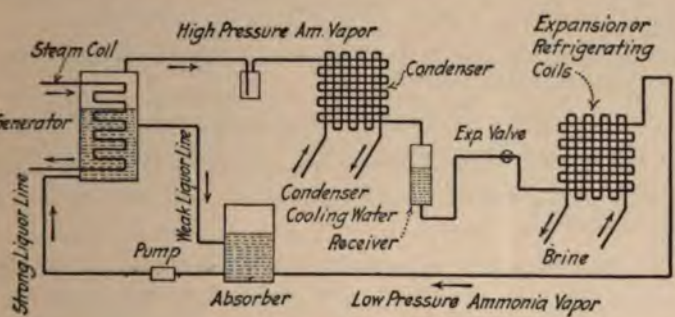

FIG. 39.

of water in which it has been dissolved. The condenser, expansion valve, and refrigerating coils are no different from those of the compression system. The ammonia vapor, after evaporation in the refrigerating coils, is conducted to the absorber, where it goes into solution in water at low pressure. This solution, called the strong liquor, is pumped into the generator, where the high pressure prevails, and the cycle is begun over again. In the meantime the water in the generator, from which the ammonia has been driven off, now called the weak liquor, is conveyed to the absorber, to receive its charge of ammonia again.

The pressures of the system must be similar to those of Fig. 37, since they depend upon conditions of natural water temperature, and load on the plant, and not upon any par-

ticular system. The generator, absorber, and strong liquor pump replace the compressor of the compression plant. The absorption system has the advantage of requiring no moving machinery, except the small liquid pump. On the other hand, the system is more difficult to control. There are many auxiliaries necessary for the operation of the plant other than those shown in the sketch of Fig. 39. On the whole, many more installations of the compression system will be found than of the absorption system.

Problems

1. Referring to Fig. 37, suppose the following observations of an ammonia refrigerating plant have been recorded:

At *A*, the pressure is 20 pounds abs.; the temperature, 26°.

At *B*, the pressure is 190 pounds abs.; the temperature, 216°.

At *C*, *D*, *E*, and *H*, the temperatures are 198°, 85°, 80° and 15°, respectively.

(*a*) From these data, and with the help of the Tables of Properties of Ammonia, construct to scale, a Temperature-entropy diagram similar to Fig. 38, to represent the changes of state of the ammonia.

(*b*) For convenience, assume the compression line (Fig. 38) to be straight. Determine the following heat quantities per pound of ammonia.

Heat removed from ammonia by compressor cylinder jacket.

Heat removed from ammonia by " radiation " from pipe line from compressor to condenser.

Heat removed from ammonia in condenser.

Heat removed from ammonia by radiation from liquid pipe line and receiver on the way from condenser to expansion valve.

Heat received by ammonia in expansion coils, from brine.

Heat received by ammonia by warming effect of atmosphere on vapor pipe line from expansion coils to compressor.

Heat equivalent of work done by the compressor upon the ammonia.

(*c*) If 843 pounds of ammonia are circulated per hour, what is the refrigerating capacity of the plant in terms of the unit of refrigeration, viz., tons per twenty-four hours?

For the solution of the above problem, the following data from

the ammonia tables may be used in case a copy of the tables themselves is not available:

Pressure—Pounds Abs.	20	190
Temperature at Saturation...	−15.9	92.7
Specific volume at 100 per cent.	13.45	1.58
Heat of liquid (h).....	− 50.3	68.6
Latent heat of vaporization...	584.3	490.9
Entropy of the liquid.....	−.1075	.1296
Entropy of evaporation.....	1.3168	.8887
Specific heat of sup. am. (cp).	.554	.658
Specific heat of liquid am.....	1.095	

2. If it is desired to maintain a difference of 30° in temperature between the cooling water supplied to a condenser, and the refrigerating agent, what compression pressure must be maintained for ammonia, carbon dioxide, and sulphur dioxide, respectively, if cooling water is furnished at 75°? At 40°?

CHAPTER IX

THE COMPRESSION AND EXPANSION OF PERMANENT GASES. CONDITIONAL RELATIONS

74. A Compressed Air System.—Fig. 40 is a diagrammatic representation of the essential elements of a compressed-air plant. It consists of a compressor, which takes in air at A, at atmospheric conditions of pressure and temperature (14.5 pounds and 70°), and delivers it at B at a higher pressure, for example 100 pounds absolute as indicated. The air is delivered to a receiver near the compressor. The utilizer or engine, located usually at a considerable distance from the compressor, spends the energy of the compressed air, and discharges it as exhaust, back into the atmosphere.

The air will be delivered from the compressor at B, at a higher temperature than 70°, by virtue of the work done upon it. But since it pauses for a time in the receiver, and must be conveyed through pipes of considerable length to the utilizer, it will be cooled, by conduction of its heat to the surrounding air, to a temperature nearly if not quite as low as that of the atmosphere itself. In the figure it has been assumed that it has been cooled back to 70° by the time it has arrived at C, the entrance to the air engine.

A compressed-air plant is not a heat engine in the same sense that the steam-power plant is a heat engine. Its purpose is to transmit mechanical energy from one point to a more or less distant point or points, not to derive mechanical energy from the primary source of energy, viz., heat. But the air in the compressed-air plant performs the same function as does the steam in the steam plant. It is a working substance or energy vehicle, which receives energy

here and disposes of it there, and is subjected to heating effects at some points and cooling effects at others. It is therefore amenable to the same general treatment as was applied to steam. Being a permanent gas, however, as distinguished from a vapor, it will be necessary to study air in the light of the properties of permanent gases, with due respect to its own peculiar properties.

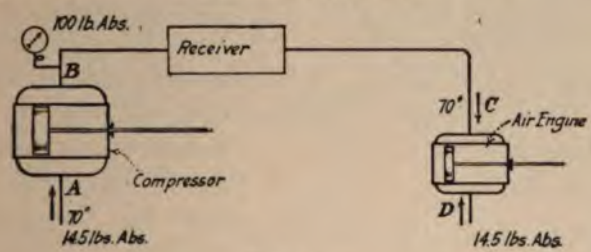

FIG. 40.

75. Distinction between a Permanent Gas and Vapor.—Air, hydrogen, oxygen and nitrogen are the most common examples of the so-called permanent gases. The term "permanent gas" was applied in the early period of the study of thermodynamics, when it was thought that such gases as were so named could not be liquefied. It is now known that all gases of which we have any knowledge can be liquefied if reduced to a sufficiently low temperature. The temperatures at which the four gases named above will liquefy, at atmospheric pressure are:

Air	−312° F.
Hydrogen	−422° F.
Oxygen	−296° F.
Nitrogen	−316° F.

It appears, therefore, that the only essential difference between a permanent gas and a vapor is that in the case of the former, our ordinary temperature range does not extend low enough to bring it near the liquid state, while in the case of the latter, our ordinary temperature range includes the possibility of either the liquid or gaseous state.

Fig. 41 is a temperature-entropy diagram for a permanent gas, carried low enougn in temperature to include its liquid state. ABCDE is a constant pressure line at 14.7 pounds. If nitrogen is considered, for example, the evaporation line BC is at the temperature $-316°$ F. or $144°$ F. absolute. The section CDE is a superheat line, of which the portion DE lies within our ordinary range of temperature. Nitrogen is familiar as a permanent gas, but it is really a highly superheated vapor. If our usual temperature range were included between $100°$ and $400°$ abs., then nitrogen would be known as a vapor, and would be treated in exactly the same manner that steam and ammonia are treated.

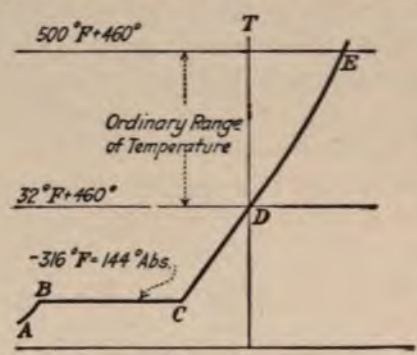

Fig. 41.

76. The Compression of Air.—The action of the compressor of Fig. 40 is represented on the PV diagram in Fig. 42. NA is the suction line; AD is the compression line; and DM is the delivery into the pipes and receiver.

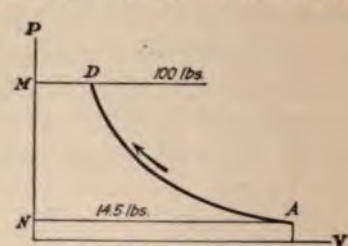

Fig. 42.

During the compression, AD, work is done upon the air by the piston, the pressure increases, the volume decreases, and the temperature tends to rise. At the same time heat may be absorbed by or rejected from the air, or there may be no heat interchange whatever. The possibility of heat absorption by the air during compression, from a heat source, may be dismissed at once, in a general discussion, since it will soon be apparent that it is never desirable to have such action, and the natural tendency

is for the air to lose heat through the walls of the cylinder. This loss of heat may be encouraged to a great extent by the use of a water jacket,the limiting condition being the maintenance of the air at the same temperature throughout compression.

Two limits are therefore set, with respect to the rejection of heat from the air; the first is the case of no loss of heat during the operation, viz., adiabatic compression; the second is the maximum rejection of heat to cause a constant degree of temperature, viz., isothermal compression. Consideration will be given to the two limiting con-

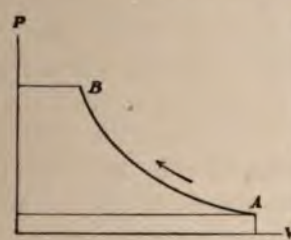

FIG. 43.

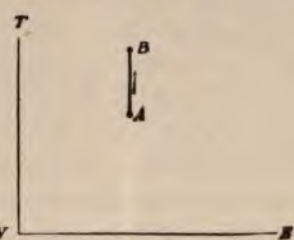

FIG. 44.

ditions first, after which an intermediate or approximately actual condition of compression, will be taken up.

77. Adiabatic Compression.—For adiabatic compression, the operation will be represented on the PV plane, Fig. 43, by the line AB, which is not, or may not be the same as the line AD of Fig. 42, which represents the general case of compression. On the TE plane, Fig. 44, the path is AB—an adiabatic constant entropy line. The direction is upward, because the temperature rises during compression if no heat is withdrawn.

At state A the pressure and temperature of the air are known by direct observation; and at state B, the pressure is known by specification. But the specific volume is not yet known at either A or B, nor is the temperature at B. It is therefore obvious that some information must now be sought concerning the properties of air, before further

progress can be made in the consideration of the energy quantities involved.

78. Relation among the Properties of Gases, Boyle's and Charles' Laws.—All knowledge of the properties of gases must rest fundamentally upon experimental evidence. For steam and other vapors, the relations among the several properties, deduced from experiment, are quite complex, and for that reason tables of their properties are constructed, thus avoiding the use of cumbersome equations in the solution of problems. On the other hand, for gases, the laws of relationship are quite simple, and it is easier to employ the equations expressing them than to resort to tables of properties. The laws of Boyle and Charles express the two fundamental relations for gases.

The first was discovered by Boyle in 1662. It states that: *at constant temperature, the volume of a given weight of gas is inversely proportional to the absolute pressure.* This law was also discovered independently a few years later by Marriotte, whose name is sometimes given to it.

The other law was discovered by Charles in 1787, but was not made public by him at the time. About 1802, Gay-Lussac published it, giving Charles the credit for its discovery however. It is sometimes spoken of as Gay-Lussac's Law. It states that: *with pressure constant, the volume of a gas varies directly as the absolute temperature.*

The statement of Charles' Law is equivalent to saying that the coefficient of expansion of a gas is constant; that is, its change of volume per degree of temperature change, is constant. If, for example, the temperature of a gas is changed from 32° F. to 33° F. or from 32° F. to 31° F. its volume will increase or decrease $\frac{1}{491.6}$ of the original, no matter what the pressure is, so long as the pressure is held constant. If its temperature is changed from 32° F. to 52° F., or from 32° F. to 12° F. its volume will increase or decrease $\frac{20}{491.6}$ of the original.

79. Graphical Representation of Charles' Law—Absolute Zero of Temperature.—Fig. 45 represents temperatures on the vertical, and volumes on the horizontal scale. The section AB of the line ABC may be said to lie within the ordinary temperatures of observation, and it is a straight line, according to Charles' Law. If the line AB be continued straight, it would cut the vertical axis at C, a distance of about 460° below the zero of the Fahrenheit scale. This gives the first conception of the absolute zero of temperature. It is the temperature at which volume of a gas would apparently disappear. The line AB would represent the result of an experiment at some pressure P_1, held constant while temperature and volume were varied. The line DE would correspond to another and higher pressure, P_2, on the same weight of gas.

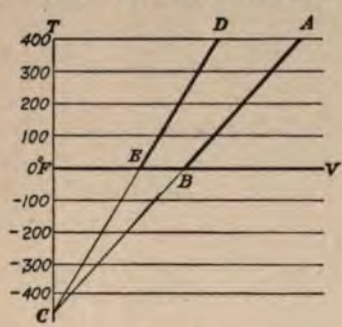

FIG. 45.

Charles' Law also expresses the relationship between temperature and pressure when the volume is kept constant, as follows: *With volume constant, the absolute pressure varies directly as the absolute temperature.*

80. Characteristic Equation of a Gas.—From Boyle's Law, $P_1V_1 = P_2V_2 = P_3V_3 = \ldots PV = \text{Constant}$.

From Charles' Law $\dfrac{V_1}{V_2} = \dfrac{T_1}{T_2}$ or $\dfrac{V_1}{V} = \dfrac{T_1}{T}$

also $\dfrac{P_1}{P_2} = \dfrac{T_1}{T_2}$ or $\dfrac{P_1}{P} = \dfrac{T_1}{T}$

The two may be combined to give the very useful *characteristic equation*, viz., $PV = WRT$.

In Fig. 46, the initial state of a pound of gas is P_1, V_1, and T_1 at (1), and it is to be changed to state (2), with values of pressure, volume, and temperature, all different from those at state (1).

First, suppose the gas to change to an intermediate state (X) at constant pressure; then from state (X) to state (2) at constant volume.

Then

$$P_x = P_1 \text{ and } V_x = V_2$$

From Charles' Law, for the first step:

$$\frac{T_x}{T_1} = \frac{V_x}{V_1} = \frac{V_2}{V_1} \qquad T_x = T_1 \frac{V_2}{V_1}$$

Also from Charles' Law, for the second step:

$$\frac{T_x}{T_2} = \frac{P_x}{P_2} = \frac{P_1}{P_2} \qquad T_x = T_2 \frac{P_1}{P_2}$$

As a result

$$T_1 \frac{V_2}{V_1} = T_2 \frac{P_1}{P_2}$$

or

$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \ldots = \frac{PV}{T} =$ a constant, which is designated by R.

Whence $PV = RT$.

The same result will be reached if the gas be supposed to change from (1) to (X) at constant temperature, and from (X) to (2) at constant volume; or from (1) to (X) at constant volume, and from (X) to (2) at constant pressure.

The assumption of one pound weight of gas was made at the outset. Hence V refers to one pound and is the specific volume. In case the weight considered is other than unity, then the symbol V should allude to the volume of the entire amount, and the expression becomes:

$$PV = WRT$$

where W is the weight of gas considered,

P is the pressure in pounds per sq. ft. abs.;

and V is the volume of the weight W, in cu. ft.

81. The Value of R.—If the specific volume of a gas is known for any one condition of pressure and temperature, the values of P, V, and T can be inserted, and R can be found. For air, the specific volume at 14.7 pounds abs. and 32° F. is known, from experiment, to be 12.39.

$$R = \frac{(14.7 \times 144) \times 12.39}{* 491.6} = 53.34$$

Knowing R, the specific volume of air can be computed for any pressure and temperature condition.

Each different gas has its own particular value of R.

82. Perfect Gas.—Actual gases are found to deviate slightly from strict adherence to the laws just discussed, and the characteristic equation derived from them. In Fig. 45, the lines ABC or DEC would not be quite straight for any actual gas. The term "Perfect Gas" is employed to express the conception of a gas that would conform exactly to the laws of Boyle and Charles.

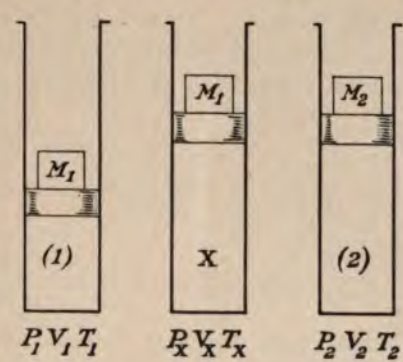

FIG. 46.

The absolute zero of temperature is not therefore exactly established by the method indicated in Fig. 45, but is located by applying as closely as possible corrections due to the imperfections of air, or any other gas, as measured by the standard of a perfect gas.

83. The Initial Specific Volume at A (Fig. 43), the Beginning of Compression.—Having now worked out the equation

$$PV = RT$$

another step can be made in the solution of the air com-

* In establishing the value of R, the more accurate value of the absolute temperature is used instead of the approximate 492.

pressor problem. The air entering the compressor (Art. 77, Fig. 43) is at state A, 70° F. and 14.5 pounds. pressure.

Whence $P_1 = 14.5 \times 144$, and $T_1 = 70 + 460$ and

$$V_1 = \frac{53.34 \times 530}{14.5 \times 144} = 13.55$$

84. The Equation of the Adiabatic.—Pressure, volume and temperature are now known at the beginning of the compression. Before the volume and temperature at B (Fig. 43), the end of compression, can be determined, the equation of the constant entropy adiabatic curve must be derived.

In making this derivation, it is necessary to employ a device similar to that used in Art. 80, in the derivation of the characteristic equation. Imagine the air to be first heated at constant volume to X, Fig. 47, and then cooled at constant pressure to B. It is desired to show this same thing on the temperature-entropy diagram. But before this can be done intelligently, it is necessary to digress again from the main thought in order to discuss the representation of constant volume and constant pressure lines on the TE plane.

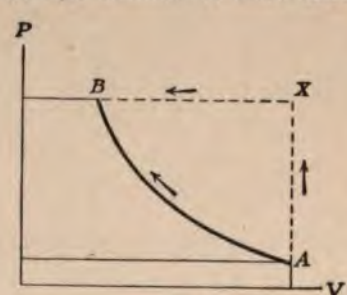

FIG. 47

85. Specific Heat at Constant Volume and at Constant Pressure.—Specific heat, in general, is defined as the amount of heat necessary to change the temperature of a unit weight of a substance one degree. Or, more definitely, with respect to English units, specific heat is the number of B.T.U. necessary to change the temperature of one pound of a substance 1° F.

The specific heat of a solid or a liquid is always the same, for any temperature. That is, it always requires the same

number of heat units to change the temperature of a pound of water from 60° to 100°, no matter if there is a constant pressure of 100 pounds per sq. in., or 1000 pounds, or a variable pressure. But the case is entirely different for a gas. The application of heat to a gas induces such an increase in the activity of its molecules as to make the gas eager to push out and occupy much more space. If its confining walls yield gradually to the increased activity of the gas, then the gas does work. In such a case, all the heat that has been supplied does not appear in temperature effect. A part of it has passed on through the gas, and appears as mechanical energy. This is represented by the energy equation:

$$H = \overline{W}c(T_2 - T_1) = (S_2 - S_1) + \frac{W}{J} \qquad [1]$$

where

H is the heat supplied;

$\overline{W}$ is the weight of gas considered;

c is the specific heat;

$(S_2 - S_1)$ is the heat that remained in the gas;

$\frac{W}{J}$ is the heat that passed on through the gas to do external work;

J is the relation between foot-pounds and B.T.U. viz., 778.

It is obvious that the quantity of heat necessary to change the temperature of a given weight of gas one degree may have an infinite number of values, depending upon how much work the gas is allowed to do during the operation.

The two cases of special interest are:

Application of heat at constant volume

and

Application of heat at constant pressure.

At constant volume, no work is done and all the heat supplied remains in the gas.

$$\frac{W}{J}=0 \qquad [2]$$

and

$$\overline{W}c_v(T_2-T_1)=S_2-S_1 \qquad [3]$$

where c_v = specific heat at constant volume.

The quantity (S_2-S_1) is that part of the entire heat supplied that remains in the gas. It is the increase of intrinsic energy. No matter how much of the heat energy supplied may be applied to doing work, the measure of the increase of intrinsic energy, (S_2-S_1), is the expression $c_v(T_2-T_1)$.

At constant pressure

$$\overline{W}c_p(T_2-T_1)=(S_2-S_1)+\frac{W}{J} \qquad [4]$$

c_p is the specific heat of the gas when heated at constant pressure.

The value of (S_2-S_1) is $\overline{W}c_v(T_2-T_1)$

also

$$\frac{W}{J}=\frac{P(V_2-V_1)}{J}=\frac{P_2V_2-P_1V_1}{J}=\frac{R}{J}(T_2-T_1) \qquad [5]$$

From the values just written the useful relation among c_p, c_v, R, and J is derived, viz.,

$$c_p-c_v=\frac{R}{J} \qquad [6]$$

86. Constant Volume and Constant Pressure Lines on the *TE* Plane. — In Fig. 48 take point A as representing some initial state of a gas. Now let the gas be heated at constant volume, from T_1 to T_2. Since heat must be supplied, the line representing the change of state will pass in a direction to the right of A in order to cover an area which will represent heat supplied. The general direction of the path will be AB. The heat supplied is A_1ABB_1.

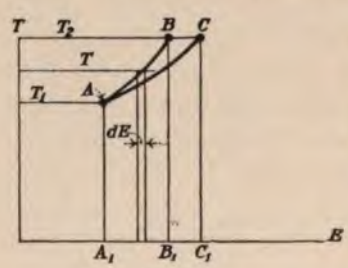

FIG. 48.

The area of the differential strip $= c_v dt$ and also $T dE$.

Hence, $$T dE = c_v dT$$

$$dE = c_v \frac{dT}{T}$$

and $$E_B - E_A = c_v \int_{T_1}^{T_2} \frac{dT}{T} = c_v \log_e \frac{T_2}{T_1}$$

which is the equation of the line AB.

When heat is supplied at constant pressure, starting from the same initial state A, the general direction of the path of change AC is the same as AB. But it will take more heat to raise the gas to the temperature T_2. The area A_1ACC_1 is greater than A_1ABB_1. The equation of the line AC is

$$E_C - E_A = c_p \log_e \frac{T_2}{T_1}$$

c_p is larger than c_v; consequently, C is located farther to the right than B, and the general conclusion may be stated that on the temperature-entropy plane, constant volume lines are steeper than constant pressure lines.

87. Values of c_p, c_v, k, R, and V for some Gases.—The relation $\frac{c_p}{c_v}$ is one of great importance. For convenience, this ratio is represented by the symbol k in this book.

$$c_p \div c_v = k$$

The values of c_p, c_v, k, R and specific volume, for the most part important gases, are given in the following table:

Gas.	c_p	c_v	k	R	Specific Volume at 14.7 pounds and 32°
Air	.241	.172	1.40	53.34	12.40
Nitrogen	.247	.176	1.40	54.99	12.79
Oxygen	.217	.155	1.40	48.25	11.22
Hydrogen	3.42	2.44	1.40	765.86	178.06

88. Derivation of the Equation of the Adiabatic, PV^k = Constant.—Returning now to the main line of thought, upon which the next objective is the establishment of a relation between the known properties at A, Fig. 43, Art. 77, and the unknown properties at B, the path of the change is represented by AB, Fig. 49. As was suggested in Art. 84, it is proposed to assume a change from A to B by first passing from A to some point, X, by way of a constant volume line AX; and from X to B by a constant pressure line XB. Inasmuch as the aim now is to establish a relation of conditions or properties between A and B, and not an energy relation, it is permissible to get from A to B by any route whatever.

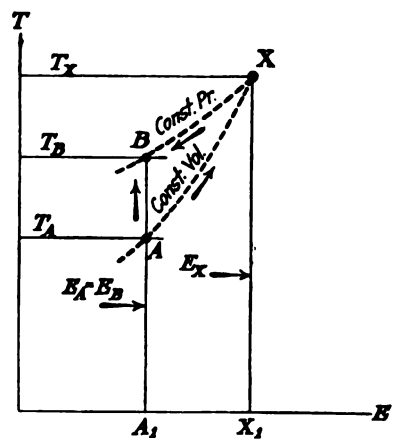

FIG. 49.

$$E_X - E_A = c_v \log_e \frac{T_X}{T_A} \qquad [1]$$

also

$$E_X - E_B = E_X - E_A = c_p \log_e \frac{T_X}{T_B} \qquad [2]$$

Whence,

$$c_p \log_e \frac{T_X}{T_B} = c_v \log_e \frac{T_X}{T_A} \qquad [3]$$

But

$$\frac{T_X}{T_B} = \frac{V_X}{V_B} = \frac{V_A}{V_B} \qquad [4]$$

And

$$\frac{T_X}{T_A} = \frac{P_X}{P_A} = \frac{P_B}{P_A} \qquad [5]$$

Whence

$$c_p \log_e \frac{V_A}{V_B} = c_v \log_e \frac{P_B}{P_A} \qquad [6]$$

$$\frac{c_p}{c_v}\log_e\frac{V_A}{V_B}=\log_e\frac{P_B}{P_A} \qquad [7]$$

$$\frac{c_p}{c_v}=k \qquad [8]$$

And

$$\left(\frac{V_A}{V_B}\right)^k=\frac{P_B}{P_A} \qquad [9]$$

Or

$$P_AV_A^k=P_BV_B^k=PV^k=\text{constant} \qquad [10]$$

which is the equation of a constant entropy adiabatic line.

89. Final Specific Volume after Adiabatic Compression.—Recalling again the air compressor of Art. 77; it is now possible to determine the final specific volume at B, Fig. 50 by means of the equation just derived.

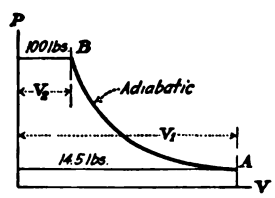

FIG. 50.

$$P_2V_2^k=P_1V_1^k$$

For air, the value of k is 1.40.

$$100\times V_2^{1.40}=14.5\times13.55^{1.40}\text{ (see Art. 83).}$$

Whence $$V_2=3.41.$$

It is obvious that the equation is applied similarly for an adiabatic expansion.

90. Final Temperature after Adiabatic Compression.—Having now the final pressure and specific volume of the air, it is easy to calculate the final temperature by the relation

$$P_2V_2=RT_2. \qquad [1]$$

$$T_2=\frac{P_2V_2}{R}=\frac{(100\times144)\times3.41}{53.34}=926^\circ\text{ F. abs.} \qquad [2]$$

or 466° F.

A direct relation between temperatures, in terms of either the pressures or volumes, can be easily derived and is frequently useful.

34515

$$P_1V_1^k = P_2V_2^k \qquad [3]$$

$$\left(\frac{V_1}{V_2}\right)^k = \frac{P_2}{P_1}, \text{ or } \left(\frac{V_1}{V_2}\right) = \left(\frac{P_2}{P_1}\right)^{\frac{1}{k}} \qquad [4]$$

Also

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}, \text{ or } \frac{T_2}{T_1} = \frac{P_2V_2}{P_1V_1} = \left(\frac{P_2}{P_1}\right)\left(\frac{V_2}{V_1}\right) \qquad [5]$$

Combining

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^k \frac{V_2}{V_1} = \left(\frac{V_1}{V_2}\right)^{k-1} \qquad [6]$$

or

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)\left(\frac{P_2}{P_1}\right)^{-\frac{1}{k}} = \left(\frac{P_2}{P_1}\right)^{1-\frac{1}{k}} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \qquad [7]$$

The expressions for $\frac{T_2}{T_1}$ just derived are convenient to use in cases where it is desired to ascertain the final temperature after adiabatic compression or expansion, without first determining both the final pressure and the final volume.

91. Isothermal Compression.—The adiabatic or constant entropy change represents one extreme or limit with respect to the withdrawal of heat during compression, while the constant temperature or isothermal operation represents the other. (See Art. 76.)

With temperature constant, Boyle's law furnishes at once the equation of the line on the PV plane.

$$P_1V_1 = P_2V_2$$

The final specific volume would be

$$V_2 = \frac{14.5 \times 13.55}{100} = 1.96 \text{ cu ft.}$$

The final temperature is 70° F., unchanged from the initial temperature.

92. Polytropic Changes.—The general form of the isothermal equation is similar to that of the adiabatic.

$$PV^1 = C_1$$

$$PV^k = C_2$$

The similarity leads to the suggestion of a general equation.

$$PV^n = C$$

where n may have any value. This is called the equation of a polytropic line.

In Figs. 51 and 52, let A indicate the initial state of a gas. Through A may be drawn a family of an infinite number of lines, all having the same general equation $PV^n = C$, and differing from each other only in the value of n.

Consider only the four special cases in which one property of a gas remains constant during a change, viz., entropy, temperature, volume or pressure.

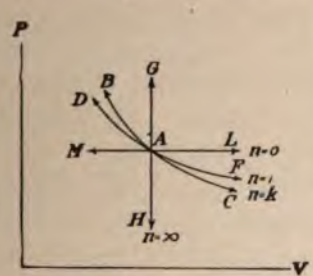

FIG. 51.

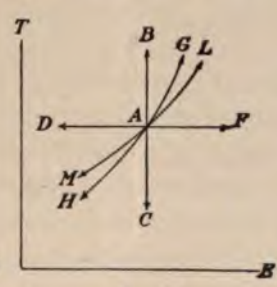

FIG. 52.

For a constant entropy compression from A, the general direction of the line is AB, as has already been discussed. The value of n is k. For adiabatic expansion, the direction is AC. To decide whether the path should proceed upward or downward on the TE plane, it is necessary only to appreciate the fact that the temperature rises during compression by virtue of the work done upon the air.

For isothermal compression, the general direction of the compression line is AD. Whether the compression line should fall below or above the adiabatic, on the PV plane, may be reasoned out thus: If the gas is compressed from the same initial state to the same final pressure, the volume after the isothermal operation will be less than after the adiabatic, on account of the cooling. Hence the isothermal

compression line falls below the adiabatic. By similar reasoning, or by a continuation of the isothermal through A, it is seen that the constant temperature expansion line AF lies above the adiabatic on the PV plane. On the TE plane the isothermal is necessarily horizontal and the direction for compression or expansion is determined by answering the question as to whether heat is withdrawn or supplied during the operation. The value of n is of course unity.

The directions taken by constant volume and constant pressure lines on the TE plane, for the case of heat supplied, are AG and AL respectively as was determined in Art. 86 For the case of cooling, or withdrawal of heat, the directions are, of course, AH and AM. On the PV plane a constant volume change is represented by a vertical line. Whether the direction is up or down, is decided by the answer to the question, is heat supplied or withdrawn, and does the pressure increase or diminish with the operation? The direction of the horizontal constant pressure line on the PV plane is settled in the same manner.

The equation of the constant volume line is

$$V_1 = V_2 = V = \text{constant},$$

which can be expressed by the polytropic form,

$$P_1 V_1^n = P_2 V_2^n = PV^n = \text{constant, if } n = \infty.$$

The equation of the constant pressure line is

$$P_1 = P_2 = P = \text{constant}.$$

Which can also be expressed by the polytropic form,

$$P_1 V_1^n = P_2 V_2^n = PV^n = \text{constant, if } n = 0.$$

For lines that fall within the angles LAH and GAM, the values of n are positive. For lines lying within the angles GAL and MAH, the values of n are negative.

93. The Actual Compression Line for an Air Compressor. — The actual compression line of a compressor

lies between the adiabatic and isothermal, and is considered to be a polytropic. The value of n is then somewhere between 1.00 and 1.40. The path of compression will be somewhat as shown by AD in Figs. 53 and 54.

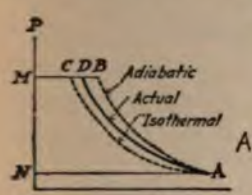

FIG. 53.

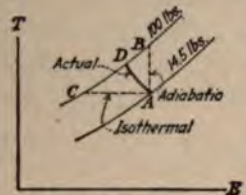

FIG. 54.

If the value of n be known either by assumption, or by the analysis of the compression line of an indicator diagram from an actual compressor, then the unknown properties, specific volume and temperature, at D can be determined by the two equations:

$$P_1 V_1^n = P_2 V_2^n$$

and

$$P_2 V_2 = R T_2$$

94. Determination of the Value of n from an Actual Compression Line.—In Fig. 55 let MN represent a polytropic line. Let the values of the ordinates be measured at the two points (1) and (2), and introduced in the equation

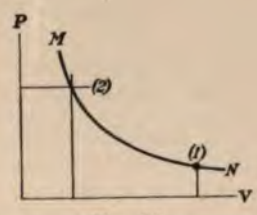

FIG. 55.

$$P_1 V_1^n = P_2 V_2^n$$

n is the only unknown, and its value is easily determined.

95. Suction and Delivery Operations of an Air Compressor.—In Fig. 53, DM represents the delivery of compressed air, and NA the filling of the cylinder with a new charge. Neither of these operations involves any change of state of the air. They are merely bodily transferences of the working substances from or to one element of the group which constitutes the system. Such operations involve energy considerations, but not conditional variations.

It is obvious that all the equations which have been derived in this chapter, with the air compressor particularly in mind, apply equally well to the expander, or air engine, where the operation is an expansion instead of a compression. Furthermore, while the discussion has centered upon air, all the deductions made are equally applicable to all permanent gas, due account being taken of the individual properties of each gas.

Problems

1. Calculate the specific volume of air at 98.4 pounds per sq. in. abs. pressure and 288° F. temperature.

2. A cylindrical tank 36×72 in. contains air at 180 pounds per sq. in. abs. pressure and 70° F. temperature. What is the weight of the air in the tank?

3. A cylindrical tank 6×36 in. contains oxygen at 3000 pounds per sq. in. abs. pressure and 80° F. temperature. What is the weight of the oxygen in the tank?

4. Determine the value of the exponent n for the polytropic line that must pass through the two points designated in each of the following cases, and indicate the direction of the lines by sketches on PV and TE planes.

(*a*) Initial pressure and volume, 10 pounds per sq. in abs. and 3 cu. ft. Final pressure and volume 120 pounds per sq. in. abs. and 11 cu ft.

(*b*) Initial pressure and volume 95 pounds per sq. in. abs. and 3.5 cu. ft. Final pressure and volume 15 pounds per sq. in. abs. and 13.5 cu. ft.

5. Referring to Fig. 53, suppose the pressure, volume and temperature at the beginning of compression, point A, to be 15 pounds per sq. in. abs., 20 cu. ft. and 70° F. temperature respectively. The compression pressure is to be 90 pounds per sq. in. abs. Construct the three lines, the isothermal, the adiabatic, and an actual compression line AD for which the polytropic, $PV^{1.20}=C$ applies. Determine by calculation values for points at 25, 40, 60 and 90 pounds per sq. in. What is the temperature at the final pressure of 90 pounds for each of the three cases?

CHAPTER X

THE COMPRESSION AND EXPANSION OF PERMANENT GASES. ENERGY RELATIONS

96. The General Energy Equation.—The general energy equation may be written:

$$H = (S_2 - S_1) + (L_2 - L_1) + \frac{W}{J}$$

where H = heat supplied in a given operation,

$(S_2 - S_1)$ = increase of sensible heat;

$(L_2 - L_1)$ = increase of latent heat;

$\frac{W}{J}$ = heat equivalent of the external work done.

In the case of permanent gases it is to be expected that no latent heat quantity is involved, since the ordinary working range of temperature is far removed from the temperature of liquefaction. Consequently the equation should simplify to

$$H = (S_2 - S_1) + \frac{W}{J}$$

97. Joule's Law.—To confirm the theory that the latent heat factor becomes zero in the general energy equation for permanent gases, the following experiment was performed by Joule (the same man who first determined experimentally the value of the mechanical equivalent of heat).

Two vessels A, and B (Fig. 56), connected by a pipe in which there was a valve V, were immersed in water contained in a tank M. The vessel A contained air compressed to about 22 atmospheres, while B was practically exhausted of air. The whole apparatus was allowed to

stand until equilibrium of temperature among the several elements and the atmospheric air was certain to have been attained. Then the valve V was opened, and the air rushed from A into B, filling both vessels at approximately 11 atmospheres pressure. The thermometer indicated that the temperature of the water after the experiment was exactly the same as before.

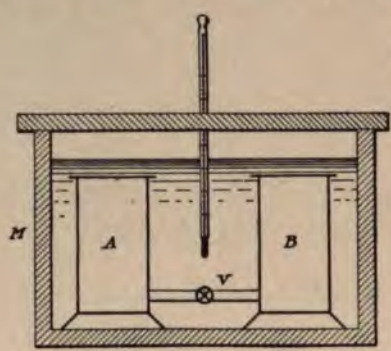

FIG. 56.

An analysis of the observations of this experiment leads to some important conclusions:

a. Since the temperature of the water was the same after the operation as before, no heat passed from the confined air to the water nor from the water to the air. Hence no heat was supplied to or rejected from the confined substance.

b. No external work could have been done by the air since there was no means of conveying it to or from the apparatus.

c. There was no change in the sensible heat of the air, since there was no temperature change.

d. If the general energy equation,

$$H = (S_2 - S_1) + (L_2 - L_1) + \frac{W}{J}$$

be considered in connection with the experiment of Joule and the deductions already made, it is seen that the quantities H, $(S_2 - S_1)$, and W, were all zero, because no net changes in their values took place. Hence $(L_2 - L_1)$ must have been zero, and

there was no latent heat effect when the pressure of the air was changed from 22 to 11 atmospheres.

The result of the experiment is thus stated in what is called Joule's Law

When a perfect gas expands without doing external work, and without taking in or giving out any heat, its temperature remains unchanged and there is no change of internal energy.

98. Deviations from Joule's Law.—In Joule's experiment, the final temperature was nearly the same as the initial temperature, but not quite, as more accurate investigations by Thompson revealed later. The deviation is very slight for the permanent gases at ordinary temperatures, and it is assumed that a perfect gas would conform exactly to Joule's Law. The deviation for actual gases grows larger the lower the temperature at which they are investigated, the gases becoming more and more influenced by the increasing proximity of their points of liquefaction. It is this principle that is made use of in the operation of the Linde liquid-air machine. At ordinary temperature ranges, actual permanent gases conform quite closely to Joule's Law, and may be treated without much error as perfect gases.

99. Working Expressions for the Energy Quantities.—The general equation for permanent gases,

$$H = (S_2 - S_1) + \frac{W}{J}$$

expresses the fundamental relation among the three energy quantities involved in a given change, viz.,

a. That which is in the form of heat received from or rejected to an outside heat reservoir (H).

b. That by which the store of intrinsic heat of the working substance is augmented or depleted ($S_2 - S_1$).

c. That which appears as external work done by or upon the working substance (W).

In order to use the equation in the solution of specific problems, it is necessary to be able to introduce for each factor, what may be termed a "working expression," which applies to the particular kind of change specified or observed; as, for example, an adiabatic, or a constant-pressure change; or some other polytropic change for which n has a definite value.

It is the purpose of this chapter to deduce some of these working expressions, and introduce them into the general equation.

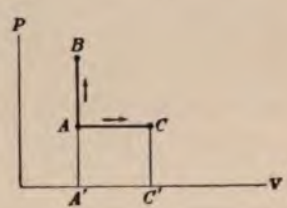

FIG. 57.

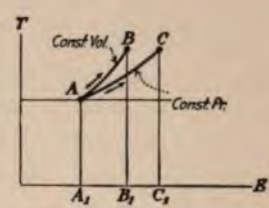

FIG. 58.

100. Constant Volume Change.—Starting from some initial state, A, Figs. 57 and 58, the lines AB represent a constant volume heating, from a temperature T_1 to T_2, of one pound of gas.

H, the heat supplied $= c_v(T_2 - T_1) =$ area A_1ABB_1, Fig. 58.

There is no work done either by the gas, or upon the gas; hence $W = 0$; and there is no area covered by the line AB on the PV plane.

Substituting now in the general equation,

$$c_v(T_2 - T_1) = (S_2 - S_1) + 0$$

All of the heat supplied appears as an increase in the store of intrinsic energy. Hence the change of intrinsic energy is measured by the expression $c_v(T_2 - T_1)$, This is always true for any substance considered as a perfect gas,

for any kind of a change. That is to say, of the entire amount of heat supplied, the portion that goes into increasing the intrinsic energy of the working substance is measured by the expression $c_v(T_2 - T_1)$, which can always be written for the general quantity $(S_2 - S_1)$.

101. Constant Pressure Change.—The lines AC, Figs. 57 and 58 represent a constant pressure heating.

The heat supplied, $H, = c_p(T_2 - T_1) =$ area A_1ACC_1, Fig. 58.

The work done is $W = P(V_2 - V_1) =$ area $A'ACC'$, Fig. 57. Substituting in the energy equation

$$c_p(T_2 - T_1) = c_v(T_2 - T_1) + \frac{P(V_2 - V_1)}{J}$$

It is from the above equation that the important relation among c_p, c_v, R and J was derived, viz.,

$$c_p - c_v = \frac{R}{J} \text{ (Art. 85)}$$

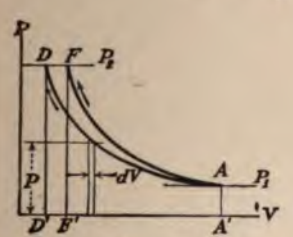

FIG. 59.

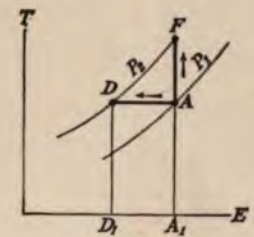

FIG. 60.

102. Constant Temperature Change—Isothermal.—AD, Figs. 59 and 60, represents an isothermal compression, and therefore a cooling operation. The amount of heat rejected, H, is not capable of being expressed directly in terms of specific heat and temperature change, because the temperature change is zero, and the specific heat is infinite. The change in sensible heat

$$(S_2 - S_1) = c_v(T_2 - T_1) = 0 \text{ because } T_2 = T_1$$

The work done upon the gas is represented by the area

$A'ADD'$ in Fig. 59, and the heat rejected, by the area A_1ADD_1 in Fig. 60.

$$A_1ADD_1 = A'ADD' \div J$$

because

$$H = O + \frac{W}{J}$$

The expression for the work done by the gas during an isothermal change can be derived by integration of the differential expression

$$dW = PdV \text{ (see Fig. 59)}$$

$$W = \int PdV$$

$$PV = P_1V_1$$

$$P = P_1V_1\frac{1}{V}$$

Hence

$$W = P_1V_1\int_{V_1}^{V_2}\frac{dV}{V} = P_1V_1 \log_e \frac{V_2}{V_1}$$

or

$$W = P_1V_1 \log_e \frac{P_1}{P_2}$$

or

$$W = RT \log_e \frac{V_2}{V_1}$$

where W = the work done, expressed in foot-pounds;

P_1 = the initial pressure, expressed in pounds per sq. ft.

V_1 = the initial volume, expressed in cu. ft.;

V_2 = the final volume, expressed in cu. ft.;

and T = the temperature, expressed in degrees F. above absolute zero.

From above

$$H = 0 + \frac{W}{J} = \frac{P_1V_1}{J} \log_e \frac{V_2}{V_1}$$

If the operation is a compression, V_2, the final volume, is less than V_1, the initial volume, and the arithmetic result

is negative, which indicates that work has been done upon the gas, and that heat has been withdrawn from the gas during the change.

On the other hand, if the operation is one of expansion, then V_2 is greater than V_1, and the arithmetic result becomes positive, indicating that work has been done by the gas, and that heat has been supplied.

It is to be observed that the area $A'ADD'$, Fig. 59, whose value is expressed by the equation

$$W = P_1 V_1 \log_e \frac{V_2}{V_1}$$

is the work of only one operation, and that it is measured to the zero line of pressure under the line of change, and is bounded by verticals dropped through the points designated as representing the initial and final states of the operation. The pressure, P_1, in the equation, must always be absolute pressure, because the area is measured to the line of zero pressure; and since W, the work, is to be expressed in foot-pounds, P_1 must be expressed in *pounds per sq. ft.*, and V_1 and V_2 in *cu. ft.*

103. Constant Entropy Change—Adiabatic.—AF, Figs. 59 and 60, represents an adiabatic compression from a pressure P_1 at A to a pressure P_2. Work is done upon the gas by an amount proportional to the area $A'AFF'$, Fig. 59. No heat is supplied or rejected from an external source, because this is the necessary condition for an adiabatic change; hence no area is covered by the line AF, Fig. 60.

The energy equation

$$H = (S_2 - S_1) + \frac{W}{J}$$

now becomes, for the case of an adiabatic change,

$$O = c_v(T_2 - T_1) + \frac{W}{J} \text{ (for unit weight)}$$

or

$$W = -Jc_v(T_2 - T_1)$$

$$W = -Jc_v\left(\frac{P_2V_2}{R} - \frac{P_1V_1}{R}\right)$$

$$= -\frac{J}{R}c_v(P_2V_2 - P_1V_1)$$

But

$$\frac{R}{J} = c_P - c_v \text{ (Art. 85)}$$

whence

$$W = -\frac{c_v}{c_P - c_v}(P_2V_2 - P_1V_1)$$

$$= -\frac{P_2V_2 - P_1V_1}{k-1}$$

or

$$W = \frac{P_2V_2 - P_1V_1}{1-k} \text{ (see Art. 104)}$$

which is the most convenient expression to use in determining the work done during an adiabatic change. If the operation is a compression, then P_2V_2 is larger, numerically than P_1V_1, and the answer will be negative. On the other hand, for an expansion, the numerator is negative, as well as the denominator, so that the final answer is positive, indicating that work is done *by* the gas.

The expression for work under an adiabatic can also be derived by integration in a manner exactly similar to that employed in deriving the equation for work under an isothermal. The procedure is illustrated in the next paragraph, where the expression for work under a polytropic is derived.

104. Polytropic Change.—Consider a polytropic change in which the pressure, volume, and temperature, change from P_1, V_1, and T_1, to P_2, V_2, and T_2. Let c represent the specific heat. Then in the general equation

$$H = (S_2 - S_1) + \frac{W}{J}$$

$$H = c(T_2 - T_1) \text{ (for unit weight)}$$

and

$$(S_2 - S_1) = c_v(T_2 - T_1)$$

The expression for W can be obtained by integration

$$W = \int P dV$$

$$P_1 V_1^n = PV^n$$

$$P = P_1 V_1^n \frac{1}{V^n}$$

Whence

$$W = P_1 V_1^n \int_{V_1}^{V_2} \frac{dV}{V^n} = P_1 V_1^n \int_{V_1}^{V_2} V^{-n} dV$$

$$= \frac{P_1 V_1^n}{1-n} (V_2^{1-n} - V_1^{1-n})$$

$$= \frac{P_1 V_1^n V_2^{1-n} - P_1 V_1^n V_1^{1-n}}{1-n}$$

But

$$P_1 V_1^n = P_2 V_2^n$$

Substituting in the first factor of the numerator

$$W = \frac{P_2 V_2^n V_2^{1-n} - P_1 V_1^n V_1^{1-n}}{1-n}$$

$$= \frac{P_2 V_2 - P_1 V_1}{1-n}$$

The expression just derived is similar in form to that derived for the work under an adiabatic (Art. 103). If the polytropic takes the special form of the adiabatic, then $n = k$, and

$$W = \frac{P_2 V_2 - P_1 V_1}{1-k}$$

In the general expression

$$H = (S_2 - S_1) + \frac{W}{J}$$

all of the factors are assumed to be positive. H is heat *added;* $(S_2 - S_1)$ is *increase* in intrinsic energy; and W is work done *by* the gas. The expression for work as derived

$$W = \frac{P_2 V_2 - P_1 V_1}{1-n}$$

assumes work done *by* the gas. If a solution results in an arithmetic value with a positive sign, then the assumption is confirmed, and the gas has done work by expansion. On the other hand, if a negative sign results, then it is shown that work has been done upon the gas by compression. It is well, therefore, to write the expression for work always in the form as derived in this paragraph. The sign of the arithmetic result then becomes an index of the nature, or rather the *direction* of the operation.

105. Specific Heat—Polytropic Change.—For any change of state, the amount of heat supplied is $c(T_2-T_1)$, where c is the specific heat. When the change is polytropic, the substitution of the " working expressions " in the general energy relation, gives the following:

$$c(T_2-T_1)=c_v(T_2-T_1)+\frac{P_2V_2-P_1V_1}{J(1-n)}$$

$$c(T_2-T_1)=c_v(T_2-T_1)+\frac{R}{J(1-n)}(T_2-T_1)$$

$$c=c_v+\frac{R}{J}\left(\frac{1}{1-n}\right)$$

But

$$\frac{R}{J}=c_p-c_v. \quad \text{(Art. 85)}$$

Substituting the equivalent of $\frac{R}{J}$,

$$c=c_v+\frac{c_p-c_v}{1-n}$$

Eliminating c_p, by substituting its value

$$c_p=kc_v$$

$$c=c_v+\frac{kc_v-c_v}{1-n}$$

$$c=c_v\left(\frac{k-n}{1-n}\right)$$

The specific heat of a gas is a peculiar property, in that it represents the ratio of the entire amount of heat supplied, during any change, to the change of temperature. It wholly ignores the other effect of the heat, viz., the external work done by or upon the gas.

The specific heat of a gas receiving or rejecting heat may have values ranging all the way from zero to infinity, and may even have negative values.

For an isothermal expansion, heat is supplied without any increase of temperature. The heat produces no effect upon the property that registers effects so far as specific heat is concerned, that is to say temperature. The specific heat is said to be infinite in value.

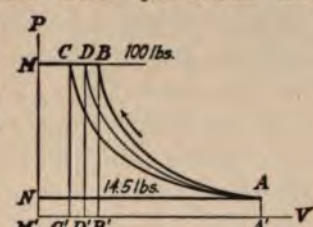

Fig. 61.

In the case of adiabatic compression, zero quantity of heat is supplied, yet there is a rise of temperature. The specific heat here is zero.

106. Energy Quantities of a Cycle.—The energy quantities that have been considered so far in this chapter have been associated with but a single operation. All actual machines or engines must operate upon a cyclic plan, and a cycle of operations will now be considered. The air compressor may be used to illustrate the principles involved, and for this purpose an ideal compressor will be assumed, working without clearance.

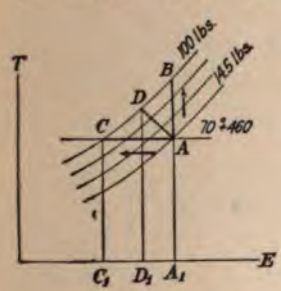

Fig. 62.

Fig. 63.

107. "Suction" of a Compressor.—Fig. 63 represents the cylinder of a compressor. Beginning with the piston at F, with zero volume behind it, the suction valve is now opened, and air at atmospheric conditions (14.5 pounds abs. and

70° F.) pushes in against the moving piston. Work is done upon the piston by the atmospheric air because of its barometric head. This operation is usually spoken of as " suction," but the term is apt to prove misleading where accurate conceptions are essential. It makes no difference whether the pressure of the atmospheric air is sufficient to move the piston against whatever resistances may be opposing its progress, or not; an amount of work is done upon the piston by an external agency, represented by the area $M'NAA'$ of Fig. 61. The temperature-entropy diagram of Fig. 62 does not picture external work; it can only show heat quantities. During the operation NA, of Fig. 61, the air that has been pushed into the cylinder has experienced no heat change. It has done work upon the piston, but only in the sense that it afforded material communication between the yielding piston and the external source of energy. Its function has been like that of the connecting-rod of an engine. It neither gains nor contributes any energy itself; it experiences no change in any of its properties. As a consequence the operation of filling the cylinder cannot be shown on the temperature-entropy diagram.

The position G, Fig. 63, is the extreme position of the piston. For one pound of air, the volume pushed in is 13.55 cu. ft. (Art. 83). The work done *upon* the piston is:

Work $M'NAA' = 14.5 \times 144 \times 13.55 = 28{,}270$ ft.-lbs. per pound of air.

108. Compression.—During compression, work is done by the piston, upon the air. Three possibilities are considered (Figs. 61 and 62).

a. Adiabatic compression along AB.

b. Isothermal compression along AC.

c. Polytropic compression along AD (assume $n = 1.25$).

For the case of the adiabatic, the specific volume at B is computed and found to be 3.41 cu. ft. and the temperature, 921° abs.

The work done is represented by the area $A'ABB'$.

$$A'ABB' = \frac{P_2V_2 - P_1V_1}{1-k} =$$

$$\frac{(100\times144\times3.41) - (14.5\times144\times13.55)}{-.40} = -52{,}070 \text{ ft.-lbs.}$$

For the case of the isothermal, the final specific volume is determined as 1.96 cu. ft. and the temperature is 70° F. or 530° abs. The work done is represented by the area $A'ACC'$, Fig. 61.

$$A'ACC' = P_1V_1 \log_e \frac{V_2}{V_1} = 14.5\times144\times13.55 \log_e \frac{1.96}{13.55}$$

$$= -54{,}590 \text{ foot-pounds.}$$

For the case of the polytropic, the final specific volume at D is found, by computation, to be 2.89 cu. ft. The work done is represented by the area $A'ADD'$.

$$A'ADD' = \frac{P_2V_2 - P_1V_1}{1-n} =$$

$$\frac{(100\times144\times2.89) - (14.5\times144\times13.55)}{-.25}$$

$$= -53{,}400 \text{ foot-pounds.}$$

During the compression of the air, its properties vary in their values, and energy interchanges take place. The operation of compression is therefore capable of being pictured on the TE diagram, and the three cases are represented by the three lines AB, AC, and AD, Fig. 62.

109. Delivery of the Compressed Air.—After the air has been compressed to the desired pressure, it must be pushed out into the receiver space. The piston must do this work, but the air which is being pushed out is again only a medium of communication between the piston, which is now the source of energy, and the reservoir of compressed

air which fills the receiver space, and acts as the recipient of the energy expended by the piston. The charge of air being expelled is crowded into this reservoir space. If the receiver has no outlet open, then the new access of energy is manifested by a rise of pressure. If the receiver is furnishing air to an air engine or other utilizer, then the new charge which has just been expelled from the cylinder into the receiver may be thought of as displacing an equal amount into the utilizer. The work of the piston of the compresser is thus transmitted directly through the entire system to the utilizer.

The delivery of the compressed air is assumed to take place at constant pressure. Hence the work of delivery is the product of the pressure times the change in volume; or, in the cases assumed, where no clearance is specified it is $-P_2V_2$, since the change in volume is $(O-V_2)$.

The work of delivery is computed for the three cases:

a. Adiabatic. $V_2=3.41$;

$$-P_2V_2=-(100\times144\times3.41)=-49{,}100 \text{ foot-pounds.}$$

b. Isothermal, $V_2=1.96$;

$$-P_2V_2=-(100\times144\times1.96)=-28{,}270 \text{ foot-pounds.}$$

c. Polytropic ($n=1.25$), $V_2=2.89$;

$$-P_2V_2=-(100\times144\times2.89)=-41{,}620 \text{ foot-pounds.}$$

The negative signs mean, of course, that the air has done negative work on the piston; that is, the piston has done work upon the air.

Obviously, the operation of delivery is not representable on the *TE* diagram, for the reasons referred to in Art. 107.

110. The Net Work of the Cycle.—Let the results of the computations of the preceding three articles be summarized in the following table. The net work of each cycle is expressed in the last item of the table on opposite page.

The negative sign, again indicates that the piston does more work *upon* the working substance, than is done *by* the working substance, which, of course, fulfills the purpose of a compressor.

	Adiabatic	Isothermal	$n=1.25$
Initial pr. (pounds per sq. in.)	14.5	14.5	14.5
Final pr. (pounds per sq. in.)	100.0	100.0	100.0
Initial specific volume, at *A*, Fig. 61	13.55	13.55	13.55
Final specific volume	3.41	1.96	2.89
Work of admission, area, Fig. 61	*M'NAA'*	*M'NAA'*	*M'NAA'*
Work of compression, area, Fig. 61	*A'ABB'*	*A'ACC'*	*A'ADD'*
Work of delivery, area, Fig. 61	*B'BMM'*	*C'CMM'*	*D'DMM'*
Net work of the cycle, area, Fig. 61	*NABM*	*NACM*	*NADM*
Work of admission, foot-pounds	+28,270	+28,270	+28,270
Work of compression, foot-pounds	−52,070	−54,590	−53,400
Work of delivery, foot-pounds	−49,100	−28,270	−41,620
Net work of the cycle, foot-pounds	−72,900	−54,590	−66,750

111. Expressions for Net Work.—Referring to Fig. 61, the net work for adiabatic compression is represented by

$$ABMN = M'NAA' + A'ABB' + B'BMM'$$

$$= P_1(V_1 - O) + \frac{P_2V_2 - P_1V_1}{1-k} + P_2(O - V_2)$$

$$= P_1V_1 + \frac{P_2V_2 - P_1V_1}{1-k} - P_2V_2$$

$$= -(P_2V_2 - P_1V_1) + \frac{P_2V_2 - P_1V_1}{1-k}$$

$$= \left(\frac{1}{1-k} - 1\right)(P_2V_2 - P_1V_1)$$

$$= \frac{k}{1-k}(P_2V_2 - P_1V_1)$$

For the case of isothermal compression, the net work, Fig. 61, is

$$ACMN = M'NAA' + A'ACC' + C'CMM'$$

$$= P_1(V_1 - O) + P_1V_1 \log_e \frac{V_2}{V_1} + P_2(O - V_2)$$

$$= P_1V_1 + P_1V_1 \log_e \frac{V_2}{V_1} - P_2V_2$$

But

$$P_1V_1 = P_2V_2$$

Hence net work, area $ACMN, = P_1V_1 \log_e \frac{V_2}{V_1}$

For the more general polytropic compression the expression for net work is obviously like that for adiabatic, except that the specific quantity k, is replaced by the general quantity, n. Hence for polytropic compression,

$$\text{Net Work} = \frac{n}{1-n}(P_2V_2 - P_1V_1)$$

These same expressions apply for expansion as well as for compression. The several factors assume values such that, in the case of compression a negative arithmetical result will be given by the solution; while for expansion a positive result will ensue.

112. Water Jacketing of Air Compressors.—Comparing the work required of an ideal compressor to compress and deliver a pound of air at a given pressure, it is evident, from the results of the examples summarized in Art. 110, that it requires much less energy when the compression is isothermal than when it is adiabatic, although it is interesting to note that in the single operation of compression it takes more work to compress isothermally. The question may now be asked: has the pound of air which has been compressed adiabatically anything more to show for the extra work, 18,310 foot-pounds, which has been expended upon it, than the pound which has been compressed isothermally? The answer depends entirely upon what happens to the

air between the time it leaves the compressor and the time it enters the utilizer. If it is transmitted without loss or gain of heat, the adiabatically compressed pound of air, because of its higher temperature and greater volume, can return more work in the utilizer, thus accounting for the excess of energy required in compressing it. In an actual plant, however, the air is cooled in traveling through the pipes and in pausing in the receiver, nearly or quite to the atmospheric temperature. In the case of the adiabatically compressed air, some mechanical energy has thus

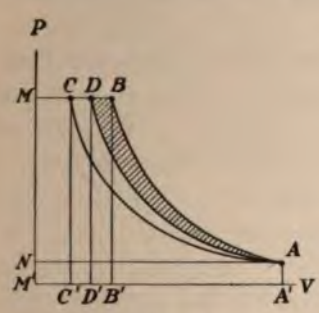

FIG. 64.

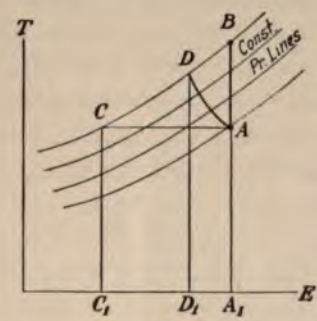

FIG. 65.

been dissipated as heat in an irreversible operation, which means an irreparable loss of available energy. In the practical general case, therefore, isothermal compression, although obviously not attainable, is the goal to be aimed at.

Water jacketing of the compressor cylinder is the practical means of accomplishing a considerable approach toward isothermal compression. In the summary of Art. 110, the polytropic $PV^n = C$, where $n = 1.25$, represents what might reasonably be expected of a water-jacketed cylinder, and shows a saving of 6150 foot-pounds over adiabatic compression.

In Figs. 64 and 65, AB, AC, and AD represent respectively, adiabatic, isothermal, and a practically attainable water-jacketed compression. The area ABD, Fig. 64,

is the saving in work due to water jacketing as compared with adiabatic compression; it represents the 6150 foot-pounds alluded to above. The area ADC is the work lost because of the imperfect action of the water jacket as compared with isothermal compression—the ideal case.

On the temperature-entropy diagram, Fig. 65, the adiabatic compression is along AB, and the subsequent cooling in the transmission system is at constant pressure along BC. The water-jacketed compression is along AD. Isothermal compression is represented by AC.

The heat that would be removed by atmospheric cooling, after adiabatic compression, is A_1BCC_1. The heat that would be removed by a perfect water jacket, resulting in isothermal compression, is A_1ACC_1. The heat removed by the actual water jacket is A_1ADD_1. The area ABD (Fig. 65) is the heat equivalent of the work saved by water jacketing. The area ADC is the heat equivalent of the work lost because the water jacketing is not perfect.

113. Interstage Cooling of Air Compressors.—Cooling the air by means of water jacketing the cylinder is only partially effective because of the limitation of wall surface through which the heat must be transmitted and the brief period of time allowed for the transmission. Further reduction of temperature can be effected by the employment of interstage cooling. The compression is split up into two or more operations or stages. In the first cylinder, the air is compressed to an intermediate pressure, and delivered into a receiver, which contains tubes through which cold water circulates. From this receiver the air passes to a second cylinder, in which the compression is completed. The interstage receiver, or cooler, is large in volume compared with the size of the cylinders, and the cooling coils present relatively large surfaces for the transmission of the heat.

The operation of a compound compressor with interstage cooler is represented in Figs. 66 and 67.

On the PV plane, Fig. 66, the admission of air to the

low-pressure cylinder is along NA. It is compressed along AF to the intermediate pressure P_r. AF lies between the adiabatic AB, and the isothermal AC. The air is then delivered from the first cylinder along FK into the interstage receiver, where it is cooled at constant pressure, suffering a reduction of volume. It enters the second cylinder along KF, but on account of shrinkage in volume, its state is now represented by G, which, being shown on the isothermal through A, means that the interstage cooler has reduced the temperature to its original value. From G, the air is compressed by the second cylinder to the final pres-

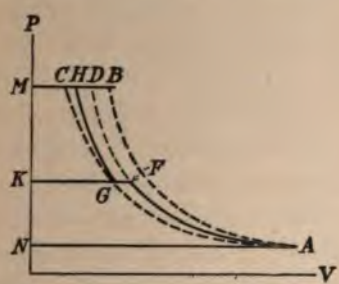

FIG. 66.

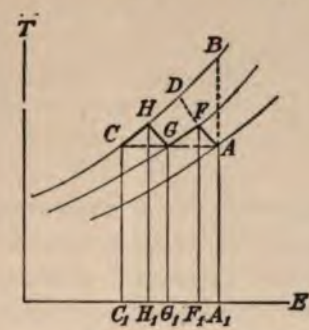

FIG. 67.

sure at H, along the line GH, after which it is delivered to the piping system or to the high-pressure receiver.

On the temperature-entropy plane, Fig. 67, AF represents compression in the first stage; FG, constant pressure cooling in the interstage cooler; and GH, compression in the second stage.

The area $AFGHB$ (both figures) represents the work saved by the employment of water-jacketed cylinders and interstage cooling, as compared with complete adiabatic compression along AB.

The area $FGHD$ (both figures) represents the work saved by the use of the interstage cooler, as compared with complete compression along AD as would be the case with a single stage water-jacketed cylinder.

On Fig. 67, the areas A_1AFF_1, F_1FGG_1 and G_1GHH_1, represent respectively the heat removed from the air by the jacket of the first stage cylinder, the intercooling receiver and the jacket of the second stage cylinder.

114. Clearance in Air Compressors.—In the preceding discussion, clearance has been disregarded in order that attention might be concentrated upon other points that were being developed. As clearance is a factor that enters into the performance of all actual compressors, its influence must now be considered. Referring to Fig. 68, NA represents the volume of one pound of air at a given initial condition as in preceding cases. Compression occurs along AD, and delivery along DM. The clearance volume is MK; hence the delivery operation ceases at K, leaving the volume MK of compressed air, still confined in front of the piston. What becomes of this residual quantity of air that the piston cannot force out?

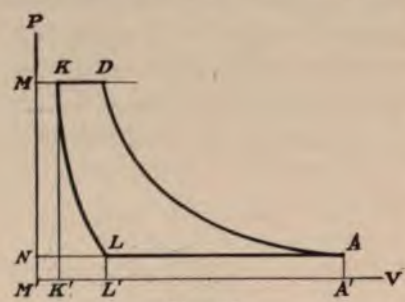

FIG. 68.

The valves of a compressor are usually automatic in their operation—that is, they open and close because of slight differences in pressure upon their surfaces. In principle they are really check valves. When the piston starts back upon its next stroke, the delivery valve closes immediately, leaving the volume MK of compressed air in the cylinder, cut off from all communication with other quantities, to re-expand along the line KL, whose law is similar to that of AD. It is not until the volume behind the piston has grown to NL that the pressure within falls to the intake pressure. Immediately thereafter the intake valve opens, and new air follows the piston, filling the cylinder at the end of the stroke by the volume LA, together with the re-expanded clearance air, of volume NL.

The compressor may be thought of as handling two kinds of air; a certain amount of new air which is drawn in, compressed, and delivered each stroke; and a certain amount of clearance air which remains in the cylinder and is compressed and expanded over and over again. The detrimental effect of clearance is that it permits the lodgment of compressed air, which, when re-expanded, occupies space that ought to be available for new air. It decreases the capacity of a cylinder by reducing the quantity of air which might be delivered in a unit time.

In Fig. 68 the distance $K'A'$ represents the piston displacement; the distance $L'A'$ represents the effective displacement. The ratio $\frac{L'A'}{K'A'}$ is called the volumetric efficiency of the cylinder. The volumetric efficiency is dependent upon the clearance; the greater the clearance, the less the volumetric efficiency.

Consider the example that has been used in illustration, in which the initial pressure is 14.5 pounds abs. and the final pressure, 100 pounds abs. For convenience assume isothermal compression. The final volume, after compression, will then be .145, or 14.5 per cent, of the original.

Assume the clearance to be 5 per cent of the piston displacement, or 4.76 per cent of the total volume at the beginning of compression, then there will be delivered only $\frac{14.5-4.76}{14.5}=.672$ or 67.2 per cent of the charge that was compressed. If the clearance is 10 per cent there will be delivered but 37.2 per cent. If the clearance is 17 per cent there will be no delivery at all.

The effect of clearance upon capacity is magnified at higher pressures of delivery or, more properly speaking, at higher ratios of compression, and minimized at the lower ratios. It is of such importance in high-pressure compressors that every precaution is taken in design and construction to reduce the clearance to a minimum.

Clearance does not much affect the amount of work to be done in compression per pound of air delivered. The clearance air returns to the piston the work which the piston does upon it. In a secondary way, energy is lost because, with clearance, a compressor would be compelled to make more strokes and would suffer more frictional effects in delivering a specified quantity of air, than would an ideal compressor having no clearance.

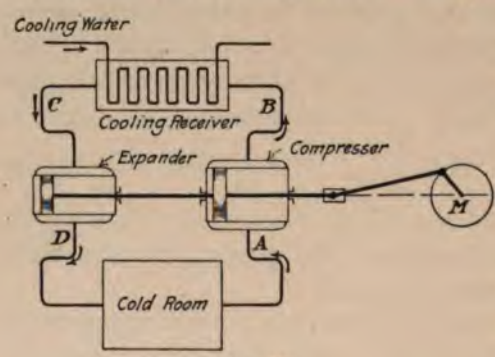

Fig. 69.

115. The Air Refrigerating Machine.—A diagrammatic sketch of an air refrigerating machine is shown in Fig. 69. The air enters the compressor at A; is delivered at B into the receiver which is also a cooler; expands in the engine or expander; and after passing through the cold room, begins its circuit over again.

The essential element of the system is the expander. In it, the working substance is to be lowered in temperature from that at C, which is the minimum obtainable by agents at natural temperature, to that at D, a temperature low enough to effect the purpose of the machine. Cooling is effected by the reduction of the intrinsic energy of the air at C, by the doing of external work. Here is seen the fundamental difference in principle between the air refrigerating machine, and the vapor refrigerating machine. In the former, lowering in temperature of the working agent must be attained by the withdrawal of energy in the form of work; in the latter, lowering of temperature is attained by merely dropping the pressure, that is, by throttling.

It is seen that the air refrigerating machine is, after all, nothing more than a compressed-air system, such as was

illustrated at the beginning of the study of gases (Fig. 40), with the exception that the circuit of the system is closed so that the same quantity of working substance is used over and over again, instead of being exhausted into the atmospheric reservoir as in the general case of Fig. 40, and with the further exception that in the refrigerating machine, the compressor and utilizer are combined into a unit mechanism for convenience. The compressed-air system, generally speaking, utilizes air as a vehicle whereby mechanical energy at one point may be conveyed to a more or less distant station; the compressed-air refrigerating machine—

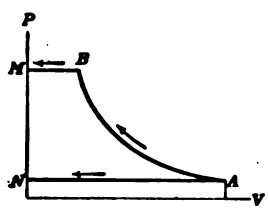

FIG. 70.

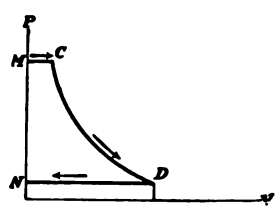

FIG. 71.

a special case of a compressed-air system—utilizes the air to obtain the thermal effects desired; the transmission of the energy of the compressor to the utilizer is incidental.

The thermodynamic action of the utilizer of any compressed-air system is the same in principle as that of the utilizer of the air refrigerating machine. Since the performance of the utilizer in the general case has not yet been discussed, the remarks that follow, in respect to the expander of the air refrigerating machine, may be considered as applicable to the utilizer of any compressed-air system. Every compressed-air system is, incidentally, a refrigerating machine. The temperature of the working medium will be lowered when work is done by it.

116. Diagrams for the Air Refrigerating Machine.—Fig. 70 represents the operation of compression on the PV plane. The compressed air, at state B, is delivered into the receiver, the line BM picturing the process. In the receiver

the air is cooled; its pressure remains constant, however, so that the withdrawal of heat results in a reduction of volume. The cooled compressed air is now admitted to the expander at M, Fig. 71, filling the cylinder to C, from which point expansion to D occurs with consequent lowering of temperature. The charge of cold air is then exhausted into the cold room, where it gradually takes up heat from the surrounding objects, cooling them, and itself becoming heated, finally re-entering the compressor cylinder along the admission line NA, Fig. 70, at a larger volume than it had upon discharge from the expander, by reason of the heat acquired in the cold room.

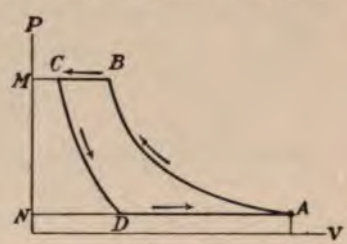

FIG. 72.

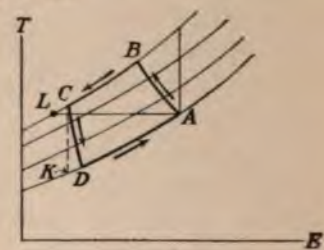

FIG. 73.

The operation of the entire machine may be represented by a combination of the diagrams of Figs. 70 and 71 into the diagram of Fig. 72. AB represents compression of the air; BC, reduction of volume by cooling in the receiver; CD, expansion in the working cylinder; and DA, growth of volume due to the warming of the cold air by the objects that it is intended to cool.

The area $NABM$ represents the work done by the compressor upon the air; and $MCDN$ the work done by the air in the expander. The work $ABCD$ must be furnished by the motor M, of Fig. 69.

In Fig. 73, the cycle of the system is shown on the TE plane. AB is compression; BC is the cooling in the receiver, at constant pressure; CD is expansion in the working cylinder; and DA is the warming of the air in the refrigerating chamber at constant pressure.

From what has already been said concerning compressors, it is obvious that the ideal compression should be isothermal, that is, along AL, to require the least work. The actual compression with water-jacketed cylinder is represented by AB. The temperature at C, after cooling, will depend upon the temperature of the cooling water and the effectiveness of the coils in the receiver as a heat exchanger. The expansion in the working cylinder should be such as will result in the maximum amount of work done, and consequent minimum exhaust temperature. It is obvious that adiabatic expansion is to be desired here. Practically, however, adiabatic expansion is unattainable, on account of the warming effect of the cylinder's environment, together with the reheating due to frictional losses. The actual expansion is represented by CD as distinguished from the adiabatic CK.

Attention is directed here again to the fundamental difference in principle between the vapor refrigeration machine, and the air refrigerating machine. In the vapor machine the lowering of the temperature of the working medium is secured by reduction of pressure by throttling; in the air machine the necessary low temperature must be secured by the withdrawal of some of the intrinsic energy in the form of work. In the vapor machine large quantities of heat are absorbed by the working medium without a rise of temperature—that is, by evaporation and latent heat; in the air machine heat can be absorbed only as sensible heat with rising temperature.

117. The Zero of Entropy for a Perfect Gas.—Frequent use has been made of the temperature-entropy plane for the delineation in a general way of states and changes of states of gases. But there has been no attempt to construct lines to scale, or represent areas in correct proportion. The chief use of a temperature-entropy diagram is to assist in relieving the mind of the necessity of holding a multitude of simultaneous conceptions. The diagram serves as a sort of shorthand record or free-hand sketch of ideas.

In order that it may be more intelligently used in this capacity, something should be said about the zero of entropy of a perfect gas. Each of the other three linear properties of gases, viz., pressure, volume, and temperature, has a definite and conceivable zero, or starting point, from which values can be measured, or upon which diagrams can be built. It is true that the absolute zero of pressure is not attainable and cannot be experienced any more than the absolute zero of temperature; yet no difficulty is experienced in conceiving of these zeros, because straight line relationships point to their location upon our usual scales of measurement.

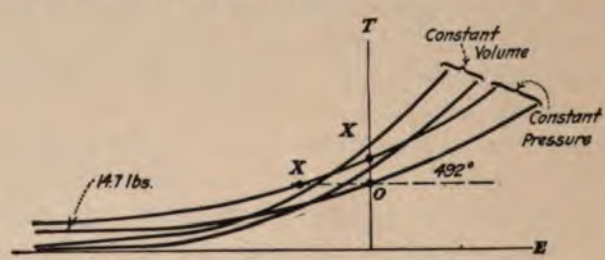

FIG. 74.

Entropy, on the other hand, has no definite beginning point. For a change of temperature from T_1 to T_2, the change of entropy can be calculated by the equation derived in Art. 86.

In Fig. 74 two constant-pressure lines and two constant-volume lines have been sketched on temperature-entropy co-ordinates.

The constant-volume lines are steeper than the constant-pressure lines, but both continue indefinitely to the left without intersecting the line of zero temperature.

The entropy of a gas may be arbitrarily considered zero when its temperature is 32° and the pressure 14.7 pounds per sq. in. abs. Upon the base line of zero temperature, erect a vertical line, and mark off a point O (Fig. 74) at 492° abs. The point O then represents the state of the

gas at 14.7 pounds pressure, 32° F., and zero entropy. Other values for temperature and entropy can then be determined for the 14.7 pounds constant-pressure line, and this line could be constructed to scale. Values for a point on some other constant-pressure line can be readily ascertained by imagining the state of the gas to pass from O, where all the properties are known, to some state X at the designated final pressure by way of either a constant temperature or a constant entropy path. By a similar process lines of constant volume of definite values can be properly related to each other, and to the constant-pressure lines, through the arbitrarily fixed point O.

Problems

1. Given 5 pounds of air at an initial condition of 50 pounds per sq. in. abs. pressure, and 80° F. For each case specified below calculate the values of the three factors, H, the heat supplied; (S_2-S_1), the change of intrinsic energy or sensible heat; and W, the external work done; as related in the energy equation,

$$H=(S_2-S_1)+\frac{W}{J},$$

also show by sketches (not to scale) on the PV and TE planes, the direction of the path of change for each case, and indicate what areas represent W and H, respectively. Be sure to make it clear whether the heat quantity is *supplied* to the air, or *withdrawn* from it; and whether work is done *by* the air or *upon* the air.

a. Constant volume heating to 400° F.

b. Constant pressure heating to 400° F.

c. Constant temperature compression to 480 pounds per sq. in abs.

d. Constant entropy expansion to 15 pounds per sq. in. abs.

e. Polytropic change, temperature rising to 400° F.; $n=1.2$.

2. An air compressor is to deliver 300 cu. ft. of free air per minute compressed to a pressure of 80 pounds per sq. in. gage pressure. The barometric pressure of the atmosphere is 29.50 in. Hg, and the temperature is 70°. Zero clearance is assumed. Illustrate all operations of the solution by sketches on PV and TE planes.

a. What theoretical horse-power would be required to drive the compressor, if the air were to be compressed adiabatically?

b. If the compressed air of (*a*) after leaving the compressor, is cooled back to 70° at constant pressure, how many heat units will have to be withdrawn per hour?

c. If the air could be compressed isothermally, what horse-power would be required, and how many heat units per hour. would have to be withdrawn from the air by water-jacketing?

d. If a compression line could be attained such that $n=1.25$, what horse-power would be saved over adiabatic compression? How many pounds of cooling water should be supplied to the cylinder jacket per hour. if the water enters at 65° and leaves at 85°?

3. A 12 in.×10 in. air compressor runs at 275 R.P.M. and is to deliver air at 115 pounds pressure absolute. If the compressor has a clearance af 3 per cent, and the compression and re-expansion lines can be expressed by the equation $PV^{1.25}=C$, what is the volumetric efficiency of the cylinder as affected by the clearance? Calculate the theoretical horse-power required for the above compressor with 3 per cent clearance; also for same compressor with zero clearance. Compare the power required in the two cases, per unit of air handled (pound or cu. ft. of free air, or 100 cu. ft. of free air).

4. It is desired to design an air compressor to compress air to 200 pounds per sq. in. abs. in two stages, from 14.5 pounds barometric pressure and 75° F.

a. Assuming isothermal compression and no clearance, determine the intermediate pressure to which the low-pressure cylinder must compress, in order that each stage shall do the same amount of work upon the air. Make the calculations on the basis of one pound of air.

b. Using the intermediate pressure determined in (*a*), and assuming the actual compression to follow the line $PV^{1.25}=C$, show what horse-power could be saved by installing an intercooler to reduce the temperature of the air at the suction of the high-pressure cylinder back to 75°, for a compressor which is to deliver 250 cu. ft. of free air per minute. How much cooling water will be needed for the jacketing of each cylinder, and for the intercooler, allowing a 20° rise in all cases? Neglect clearance. Fully illustrate all operations by use of *PV* and *TE* diagrams.

5. Make tentative calculations for an air-refrigerating machine illustrated by the diagram of Fig. 69. The maximum temperature of the air in the cold room is to be 35°, and the pressure atmospheric, (14.5 pounds). The compressor will compress the air along the line $PV^{1.2}=C.$, delivering at 110 pounds abs. The compressed air is reduced in temperature, in the cooling receiver to 85°. Expansion then takes place to the lower pressure following the equation $PV^{1.35}=C$.

a. If 40,000 B.T.U. per hour are to be removed from the cold room, how many pounds of air per minute must be circulated?

b. Disregarding clearances, what theoretical horse-power is necessary to run the compressor? How much is furnished by the expander? and how much remains to be furnished by the motor?

c. Make an energy balance table to show the amounts of energy supplied to and withdrawn from the air per minute (neglecting all

" radiation " losses) considering each operation in connection with the energy equation $H=(S_2-S_1)+\frac{W}{J}$.

6. An air or vacuum pump, such as would be used in connection with the condenser of a steam-power plant, is really an air compressor whose " suction " is below atmospheric, and whose discharge pressure is atmospheric pressure. Under what conditions of suction pressure on the pump (that is to say, vacuum in the condenser), does the pump require most power to run it? Or, stated in a slightly different way, does an air pump require more power to run it when maintaining a very high vacuum; or when just starting to clear the condenser of air, in which case the vacuum is practically zero; or when maintaining the vacuum at some intermediate value? Assume the air pump to run at the same speed in all three cases.

7. A 12 in.$\times$12 in. air compressor running at 250 R.P.M. has 2 per cent clearance and is to deliver compressed air at 105 pounds abs. pressure. Make calculations to show the effect of altitude upon the performance of this machine by computing the number of pounds of air compressed per minute and the horse-power required.

a. For location at sea level (29.92 in.).

b. For location at an altitude where the barometric pressure is 24 in. Hg.

Consider the value of n to be 1.25 for both the compression and re-expansion lines, and assume the temperature to be 70° in both cases.

CHAPTER XI

INTRINSIC ENERGY AND HEAT CONTENT. THROTTLING OF GASES

118. Net Work of the Cycle for an Air Compressor or Air Engine.—In Fig. 75 consider an adiabatic expansion from the initial state A, to the final state B. The case may be stated more concretely and practically by saying that compressed air is furnished to an air engine at 120 pounds abs. pressure, and 100° F. temperature. If the air could be expanded adiabatically to 15 pounds abs. pressure how many B.T.U. per pound of air would be available for work? It has already been shown (Art. 111) that the net work is represented by the area $MABN$, and may be computed by the expression

$$P_1V_1+\frac{P_2V_2-P_1V_1}{1-k}-P_2V_2=\frac{k}{1-k}(P_2V_2-P_1V_1)$$

Or, since the case is one of expansion and it is known at once that P_2V_2 is less than P_1V_1, the equation may be written

$$\text{Net work}=\frac{k}{k-1}(P_1V_1-P_2V_2)$$

Introducing the numerical values specified,

$$\frac{k}{k-1}(P_1V_1-P_2V_2)=46{,}800 \text{ foot-pounds or } 60.2 \text{ B.T.U.}$$

A very natural question intrudes itself at this point; since the expansion is adiabatic which conditions no supply or rejection of heat, and if all the work performed is done at the expense of the original store of intrinsic energy, why

is not the net work done equal to the decrease of intrinsic energy which is expressed by the equation

$$(S_1 - S_2) = \frac{P_1V_1 - P_2V_2}{k-1} = 33{,}425 \text{ ft.-lbs., or } 43.0 \text{ B.T.U.?}$$

In answering this question, first consider the air at state A (Fig. 75). When the compressed air is introduced into the cylinder of the air engine, as represented by MA, work is done by the medium upon the piston. But the work is not done at the expense of any of the intrinsic energy possessed by the air. The pressure at A is as high as it is in the receiver from which the air is supplied; the temperature is not any less; the specific volume is not reduced. The only change that has occurred is that a quantity of air which was in the supply reservoir, has been transferred into the cylinder. Work has been done in pushing the piston, a yielding resistance, out of the way. The supply of energy for this work came from the stored air in the receiver, and not from the particular quantity of air that is to operate the air engine in the cycle which is being considered. If the system be explored still further back toward the source of the energy, an air compressor will be found. The compressor, it will be remembered, first concentrates the air, thereby storing energy in the air itself, and then pushes out the charge. The work of the compressor in pushing out this charge is external work. The charge must find room for itself by pushing other quantities of air out of its way, until finally an equivalent charge is pushed into the utilizer. The work done in pushing the charge into the utilizer—the area $M'MAA' = P_1V_1$ (Fig. 75) is external work transferred directly from the compressor to

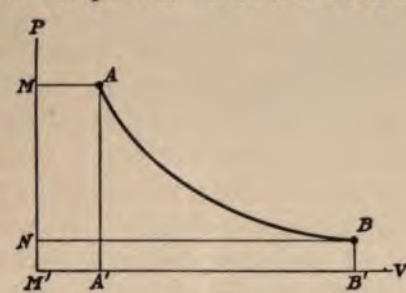

FIG. 75.

the engine. It corresponds exactly to the external work part of the heat of steam.

After the air has been expanded, it is expelled from the working chamber. It takes work to push the charge out into the atmosphere, which is really a great receiver. Whether it takes more work to expel the charge, at state B, than was done in introducing it at a state A, depends upon the relative magnitudes of the quantities, P_1V_1 and P_2V_2. As a consequence of adiabatic expansion, P_2V_2 is smaller than P_1V_1. Hence, for the total net work done upon the piston, there is a profit from the transaction of introducing and expelling the charge, to add to the work done at the expense of its intrinsic energy. For adiabatic (constant entropy) expansion the net work is k times the reduction in intrinsic energy, or the gain in work by the reduction of PV is 40 per cent of the change in intrinsic energy.

If the expansion (or compression) line AB, of Fig. 75 were an isothermal, the net work of the cycle would become:

$$\text{Net work} = P_1V_1 + P_1V_1 \log_e \frac{V_2}{V_1} - P_2V_2$$

But

$$P_1V_1 = P_2V_2$$

Hence,

$$\text{Net work} = P_1V_1 \log_e \frac{V_2}{V_1}$$

The area under an isothermal line, bounded by the verticals dropped through the terminal points A and B (Fig. 75), is equal to the area to the left of the line, bounded by the horizontals through the terminal points.

The net work of the cycle when AB is a polytropic of equation

$$PV^n = C$$

is given by the expression

$$\frac{n}{1-n}(P_2V_2 - P_1V_1)$$

which is, of course, identical in form with that of the special case of the adiabatic.

119. Heat Content of Gases.—Referring to Fig. 75, and to some of the points discussed in the preceding paragraph, the net work of the cycle, assuming adiabatic expansion, may be written

$$\text{Net work} = P_1V_1 + \frac{P_2V_2 - P_1V_1}{1-k} - P_2V_2$$

$$= \left(P_1V_1 + \frac{P_1V_1}{k-1}\right) - \left(P_2V_2 + \frac{P_2V_2}{k-1}\right)$$

Each of the expressions within the parentheses contains two factors—one expressing the work of introduction or external work, and the other the intrinsic energy, at the state represented. The quantities within the parentheses may be called the work equivalent of the *heat contents* of the gas at the two states, so that

$$JH_1 = P_1V_1 + \frac{P_1V_1}{k-1} \quad \text{and} \quad JH_2 = P_2V_2 + \frac{P_2V_2}{k-1}$$

or, generally

$$JH = PV + \frac{PV}{k-1}$$

Heat content may also be expressed in terms of temperature:

$$JH = PV + \frac{PV}{k-1}$$

$$= RT + \frac{RT}{k-1}$$

$$= R\left[1 + \frac{1}{k-1}\right]T$$

$$H = \frac{R}{J}\frac{k}{k-1}T$$

But

$$\frac{R}{J} = c_p - c_v; \text{ and } k = \frac{c_p}{c_v}$$

Substituting

$$H = \left[\left(\frac{c_p - c_v}{c_v}\right) \div \left(\frac{c_p}{c_v} - 1\right)\right]c_pT$$

Or

$$H = c_pT$$

If a truly permanent or uncondensable gas existed, the heat content at any absolute temperature T, would be the heat that was necessary to bring a pound of it up from zero absolute to temperature T at constant pressure. The heat so applied would appear partly as sensible heat in the rise of temperature, and partly as external work that had to be done by the gas in making room for itself against an environing pressure.

The heat content of a gas may thus be considered to be equal to the specific heat at constant pressure times the absolute temperature.

The term *heat content* is a misnomer when applied to gas as well as when applied to steam. It does not designate the heat which a pound of working substance "contains" at a specified state; the term *intrinsic energy* properly covers that idea. The term *heat content*, when applied to steam, alludes to the amount of heat required to create a pound of steam at a specified state, starting with water (which has relatively no volume) at the arbitrary zero of heat measurement, viz., 32°, and proceeding at constant pressure to the specified state. The term *heat content* when used in connection with gases, may be considered as the amount of heat required to create a pound of gas, starting with zero volume at zero absolute temperature, and proceeding at constant pressure to the state specified. The term has, therefore, exactly the same significance when used in connection with either vapors or gases.

When the net work of the cycle for an air engine or an air compressor is under consideration and no passage of heat to or from the working substance is assumed, the heat equivalent of the useful work is the difference between the initial and final heat contents.

$$\text{Net work of the cycle} = J(H_1 - H_2).$$

When the operation is assumed to be a constant entropy adiabatic, then the difference between the heat contents is the *available energy* put into the air by compression, or ready to do work in expansion.

When any of the operations of the cycle assumed are accompanied by the interchange of heat between working substance and outside sources, then it is preferable to analyze each individual operation with the help of the general energy equation,

$$H = (S_2 - S_1) + \frac{W}{J}$$

and summarize the quantities of each factor of the equation to arrive at the net effects for the cycle.

120. Throttling of Gases.—When air, or any gas considered as a perfect gas, is reduced in pressure, without doing any useful external work as, for example, when it is throttled in a valve, passes through an orifice, is wire drawn through restricted ports, or suffers a frictional effect while flowing in a pipe, the operation is accompanied by no change of temperature. This is a necessary conclusion from Joule's experiment (Art. 97) because, by that experiment, it was shown that there is no latent heat effect for a perfect gas, and the energy equation can be written

$$H = (S_2 - S_1) + \frac{W}{J}$$

In a throttling operation both H and $\frac{W}{J}$ are zero in value.

Hence $(S_2 - S_1)$ must be zero. Inasmuch as $(S_2 - S_1)$ is expressed in value by the term $c_v(T_2 - T_1)$, it necessarily follows that the operation of throttling is one of constant temperature, or rather, one of equality between the final and initial temperature.

Referring to Figs. 76 and 77, the point A represents the initial state of a gas at, for example, 120 pounds abs. pressure and 560° abs. temperature. Suppose the gas to be throttled to 15 pounds abs. pressure. The final temperature is the same as the initial, and the point B, representing the final state, will be found on the isothermal through A. The area under the line has no significance in either figure.

121. Expansion of Compressed Air from a Reservoir.— Referring to Fig. 78, A is a tank or reservoir containing compressed air at a pressure of P_1 pounds per sq. in. abs. and a temperature of $T°$. The tank is cut off from any supply. Now let the valve F be partially opened, allowing

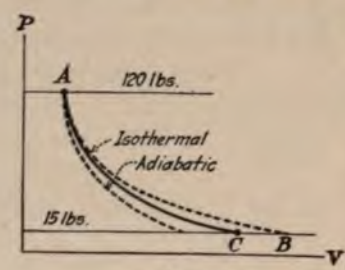

Fig. 76.

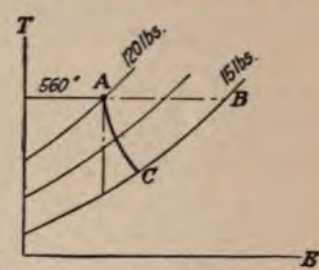

Fig. 77.

some of the compressed air to escape into the atmosphere. The pressure within the tank will of course decrease. What will happen to the temperature of the air remaining in the tank? What will be the temperature of the escaped air, before it has been influenced by commingling with the atmospheric air? What will be the average temperature of the entire mass of air originally contained in the tank?

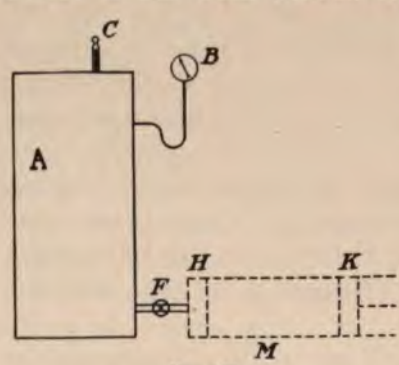

Fig. 78.

To assist in gaining a clearer conception of what occurs, imagine a cylinder M, shown in dotted lines in Fig. 78, to receive the escaping air, and to keep it separated from the atmospheric air. At the beginning of the operation, the piston is at H, and at the end it is at K, having been moved from H to K against the constant pressure of the atmosphere. The work done by the confined air in pushing the piston before it against the atmospheric pressure, is obvi-

ously the same as if the air escaped directly into the atmosphere. Work is done, in any case, by the escaping air. The energy for this work can come from no place save the original store of intrinsic energy possessed by the air in the tank. The average temperature of the entire mass of air will therefore be lowered by the operation.

122. Final Temperature of Air Remaining in the Tank.—The work of displacing the piston is really done by the air that remains in the tank. As soon as a quantity of air escapes from confinement it experiences no further loss of intrinsic energy and consequent cooling effect. Inasmuch as it is assumed that the air in the tank neither receives nor gives out any heat during the operation, the expansion is adiabatic for all of the air that remains up to a given moment. For the final pressure observed, P_t, the temperature T_t is to be computed by the adiabatic relation

$$T_t = T_1\left(\frac{P_t}{P_1}\right)^{\frac{k-1}{k}}$$

This relation holds true whether the valve F (Fig. 78) remains open, allowing complete equilibrium of pressure within the tank with that of the atmosphere, or whether the operation is arrested by the closure of the valve before P_t reaches equality with the atmospheric pressure.

123. Average Temperature of Escaped Air.—The escaped air will not be as cool as the air remaining in the tank, because it is made up of contributions from the tank, varying in temperature from the warmest air in the tank at the very beginning of the operation to the coldest, resulting from adiabatic expansion down to the termination of the operation.

Let $\overline{W}_1$ = weight of the air in the tank at initial conditions;

$\overline{W}_t$ = weight of the air in the tank finally;

$\overline{W}_e$ = weight of the escaped air;

T_1 = initial temperature of air in the tank;
T_t = final temperature of air in the tank;
T_e = average temperature of escaped air;
T' = average temperature of entire original weight of air, after the operation;

P_1 = initial pressure of air in the tank;
P_t = final pressure of air in the tank;
P_a = atmospheric pressure;

V_t = the volume of the tank;
V_e = the volume of the escaped air at P_a and T_e.

The work done in displacing atmospheric air $= P_a V_e$. The energy to perform this work comes from the intrinsic energy of the air.

Loss of intrinsic energy $= \overline{W}_1 c_v (T_1 - T')$

$$T' = \frac{\overline{W}_t T_t + \overline{W}_e T_e}{\overline{W}_1}$$

also

$$P_a V_e = \overline{W}_e R T_e$$

Substituting

$$\overline{W}_e R T_e = J \overline{W}_1 c_v \left[T_1 - \frac{\overline{W}_t T_t + \overline{W}_e T_e}{\overline{W}_1} \right]$$

$$\overline{W}_e R T_e = J \overline{W}_1 c_v T_1 - J \overline{W}_t c_v T_t - J \overline{W}_e c_v T_e$$

$$\overline{W}_e (R + J c_v) T_e = J c_v (\overline{W}_1 T_1 - \overline{W}_t T_t)$$

$$T_e = \frac{J c_v}{R + J c_v} \frac{\overline{W}_1 T_1 - \overline{W}_t T_t}{\overline{W}_e} = \frac{1}{k} \frac{\overline{W}_1 T_1 - \overline{W}_t T_t}{\overline{W}_e}$$

124. Example.—Suppose the volume of the tank of Fig. 78 to be 2 cu. ft.; the initial pressure 64.5 pounds per sq. in. abs.; and the initial temperature 80°. Let the valve F be opened, allowing the compressed air to escape until the pressure within the tank is equal to the atmospheric pressure which is assumed to be 14.5 pounds per sq. in. abs. What is the temperature of the air remaining in the

tank; the average temperature of the escaped air; and the average temperature of the entire mass of air originally confined within the tank?

$$\overline{W}_1 = \frac{P_1 V_t}{RT_1} = \frac{64.5 \times 144 \times 2}{53.34 \times 540} = .644 \text{ pound}$$

$$T_t = T_1 \left(\frac{P_t}{P_1}\right)^{\frac{k-1}{k}} = 540 \left(\frac{14.5}{64.5}\right)^{.289} = 353° \text{ temperature of air remaining in the tank}$$

$$\overline{W}_t = \frac{P_t V_t}{RT_t} = \frac{14.5 \times 144 \times 2}{53.34 \times 353} = .222 \text{ pound}$$

$$\overline{W}_e = \overline{W}_1 - \overline{W}_t = .644 - .222 = .422 \text{ pound}$$

$$T_e = \frac{1}{k} \frac{\overline{W}_1 T_1 - \overline{W}_t T_t}{\overline{W}_1} = \frac{.644 \times 540 - .222 \times 353}{1.40 \times .422}$$

$= 456°$ average temperature of escaped air

$$T' = \frac{\overline{W}_t T_t + \overline{W}_e T_e}{\overline{W}_1} = \frac{.222 \times 353 + .422 \times 456}{.644}$$

$= 420°$, average final temperature of the mass of air originally confined in the tank.

Suppose that the experiment was repeated, but instead of permitting the pressure in the tank to fall to atmospheric, let the operation be arrested when the pressure in the tank is 34.5 pounds abs.

$$T_t = T_1 \left(\frac{P_t}{P_1}\right)^{\frac{k-1}{k}} = 540 \left(\frac{34.5}{64.5}\right)^{.289} = 452°$$

$$\overline{W}_t = \frac{P_t V_t}{RT_t} = \frac{34.5 \times 144 \times 2}{53.34 \times 452} = .412 \text{ pound air remaining in tank}$$

$$\overline{W}_e = .644 - .412 = .232 \text{ pound escaped air}$$

$$T_e = \frac{1}{k} \frac{\overline{W}_1 T_1 - \overline{W}_t T_t}{\overline{W}_e} = \frac{.644 \times 540 - .412 \times 452}{1.40 \times .232} = 497°$$

Cooling could be effected by proper confinement and circulation of the escaped air. The lowering of temperature by this means is, however, only about one-half that which results from adiabatic expansion of the whole body of air.

Problems

1. Compressed air is supplied to an air turbine at 125 pounds per sq. in. abs. pressure, and 250° F. temperature. The turbine exhausts at 15 pounds abs. pressure. The temperature of the exhaust is found to be 60° F. If the turbine develops 10 B.H.P. as shown by test, how many pounds of air are used per hour?

2. An automobile tire contains 1200 cu. in. of air at 70 pounds pressure and 90° F. temperature.

(*a*) If the tire sustains a load of 750 pounds, what is the area of contact with the ground, assuming the rubber of the tire to be perfectly flexible?

(*b*) If the valve is opened for a time, and the pressure reduced to 20 pounds gage, what will be the temperature of the air remaining in the tire, if it receives no heat from the walls? What weight of air did the tire contain originally? how much escapes through the valve? and what is the theoretical average temperature of the escaped air?

CHAPTER XII

MIXTURES OF GASEOUS SUBSTANCES

125. Occurrence of Gaseous Mixtures in Engineering Work.—It is frequently the case that problems must be dealt with which involve the action not of a simple gas or vapor alone, but a mixture of two or more gases, a mixture of gas and vapor, or a mixture of two or more vapors. Examples of such cases are the charge of an internal combustion engine, where air is mixed with a gaseous fuel; the products of combustion in an internal combustion engine, or a boiler furnace; the mixture of air and vapor in the condenser of a condensing steam plant; the mixture of pure air and water vapor which constitutes the atmosphere. In the case of the atmosphere the pure air so much exceeds the vapor in weight that under most circumstances no appreciable error results from a total neglect of the moisture. However, it is of importance to be able to analyze the problem of mixtures, for it is only by this means that it can be shown whether or not it is necessary to apply the refinement in a given class of cases.

126. Weight and Volume Relations.—A mixture may be specified by expressing the quantity of each constituent either in terms of its percentage by weight or its percentage by volume. If the density, or its reciprocal, the specific volume, of each constituent is known, conversion from one expression to the other can easily be made.

Suppose the mixture, air, be considered in terms of its two chief constituents, nitrogen and oxygen. Its constitution by volume is, nitrogen 79.3 per cent and oxygen 20.7 per cent. What is its constitution by weight? The density (weight per cu. ft.) under standard conditions, of

nitrogen is .0783, and that of oxygen is .0892. (See table of Art. 130.)

Gas	Per cent by Volume.		Density.		Weight of Each in 100 Volumes of Mixture.	Per cent by Weight.
N_2	79.3	×	.0783	=	6.2084	6.2084 ÷ 8.0552 = 77%
O_2	20.7	×	.0892	=	1.8468	1.8468 ÷ 8.0552 = 23%
					8.0552	

The average density of the mixture is found by dividing the weight of 100 volumes (8.0552) by the number of volumes (100).

Average density = 8.0552 ÷ 100 = .08055.

If the constitution by weight is given, and it is desired to find the proportion of the constituents by volume, the process is reversed.

Gas	Per cent by Weight.		Density.		Volume of Each in 100 Weights.	Per cent by Volume.
N_2	77	÷	.0783	=	98.35	98.35 ÷ 124.13 = 79.3
O_2	23	÷	.0892	=	25.78	25.78 ÷ 124.13 = 20.7
					124.13	

127. Densities and Molecular Weights.—In the conversions of the preceding paragraph it is obvious that, in order to arrive at the final result, the actual value, of the densities need not be used, provided numbers are employed that are proportional to the densities. Now the molecular weights of substances are proportional to their densities, and as the molecular weights of the most common gases are easily remembered, while the densities are not, it is convenient to use the molecular weights. The problem used in illustration would then be solved in the following manner;

Gas	Per cent by Volume.		Molecular Weight.		Molecules in 100 Volumes.	Per cent by Weight.
N_2	79.3	×	28.08	=	2226.7	2226.7 ÷ 2889.1 = 77
O_2	20.7	×	32.00	=	662.4	662.4 ÷ 2889.1 = 23
					2889.1	

The molecular weight of the mixture, that is, of air, is $2889 \div 100 = 28.89$.

128. Determination of R for Mixtures.—If the value of R for a mixture can be determined from a knowledge of the constituents, it can be placed in the equation $PV = RT$, and the mixture can be treated as a simple gas.

To continue the illustration with the mixture of nitrogen and oxygen called air, the value of R for the mixture can be computed directly from the density of the mixture, which has already been found to be .08055 at 14.7 pounds pressure and 32°.

$$\text{Specific volume} = \frac{1}{.08055} = 12.40$$

$$R = \frac{14.7 \times 144 \times 12.40}{492} = 53.34$$

129. The Universal Gas Constant.—The characteristic equation $PV = RT$ holds good for any permanent gas. The equation may be written

$$\frac{R}{V} = \frac{P}{T}$$

Call the density of a gas d.

Then

$$V = \frac{1}{d}$$

and

$$Rd = \frac{P}{T}$$

Let it be imagined that a number of gases are being compared. The same pressure and temperature can be

imposed upon all the samples examined. Hence for purposes of the comparison the term $\frac{P}{T}$ is a constant, and the several gases will manifest their individual differences in characteristics by different values of R and d. For all the gases, the equation may be written

$$Rd = C_1 \text{ (a constant)}$$

Since the molecular weight, m, of any gas is proportional to the density, the equation may be written:

$$Rm = C_2 \text{ (a constant)}$$

Knowing the values of R and m for any one gas, these can be introduced, and C_2 computed

For example, for air $R = 53.34$, and m, as determined in Art. 127 is 28.89.

$$C_2 = 53.34 \times 28.89 = 1544$$

This constant, 1544, Professor Goodenough calls the "Universal Gas Constant." It is equal to the value of R for any gas, multiplied by the molecular weight of that gas. In order to determine the value of R for any gas, it is only necessary to remember the value of the universal constant, and the molecular weight of the gas—a quantity which, for the simpler elements, is usually retained by the memory without effort.

$$R = 1544 \div m$$

For a chemically compound gas, as for example carbon dioxide, the molecular weight can be found by a consideration of the chemical formula.

It should be noted that, in the case of simple gases, m refers to the molecular weight, not the atomic weight.

130. Properties of Common Gases.—Some of the properties of the more common gases are tabulated below.

SOME PROPERTIES OF COMMON GASES *

Name of Gas.	Chem. Symbol.	Molecular wt.		Specific Heat.		$k=\frac{c_p}{c_v}$	Gas Constant R.	Wt. per cu. ft. in lbs. at 32° and 14.7 lbs. per sq. in.
		Approx.	Exact $O_2=32$	c_p	c_v			
Helium	He	4	4.0	1.250	.75	1.66	386.0	.0112
Argon	Ar	40	39.9	.124	.075	1.66	38.70	.1112
Air	....	29	28.95	.241	.171	1.40	53.34	.0807
Oxygen	O_2	32	32	.217	.155	1.40	48.25	.0892
Nitrogen	N_2	28	28.08	.247	.176	1.40	54.99	.0783
Hydrogen	H_2	2	2.016	3.42	2.44	1.40	765.86	.00562
Nitric oxide	NO	30	30.04	.231	.165	1.40	51.40	.0838
Carbon monoxide	CO	28	28.00	.243	.172	1.41	55.14	.0780
Hydrochloric acid	HCl	36.5	36.45	.191	.136	1.40	42.35	.1017
Carbon dioxide	CO_2	44	44.00	.210	.160	1.31	35.09	.1227
Nitrous oxide	N_2O	44	44.08	.221	.171	1.26	35.03	.1229
Sulphur dioxide	SO_2	64	64.06	.154	.123	1.25	24.10	.1786
Ammonia	NH_3	17	17.06	.523	.399	1.31	90.50	.0476
Acetylene	C_2H_2	26	26.02	.350	.270	1.28	59.34	.0725
Methyl chloride	CH_3Cl	50.5	50.47	.24	.20	1.20	30.59	.1407
Methane	CH_4	16.0	16.03	.593	.450	1.32	96.31	.0447
Ethylene	C_2H_4	28.0	28.03	.40	.33	1.20	55.08	.0780

* Marks' Mechanical Engineers' Handbook.

131. The Specific Heat of Mixtures.—Again using the mixture of nitrogen and oxygen, that forms air, as an illustration, the determination of the specific heat of a mixture from a knowledge of the composition by weight and the specific heats of the constituents is easily shown.

Gas.	Per Cent by Weight.		Specific Heat (c_p)	
N_2	77.0	×	.247	= 19.02
O_2	23.0	×	.217	= 4.99
				24.01

Specific heat of mixture = 24.01 ÷ 100 = .240.

132. The Variation of Specific Heats.—The values of specific heats given in the table of Art. 130 may be considered to apply only at ordinary temperatures. For extreme temperatures, such as are met with in furnaces and internal

combustion engines, the true values of specific heats deviate too far from those given in the table to warrant their use.

For those gases that follow quite closely the laws of perfect gases, the specific heat is independent of the pressure but may vary with the temperature. That there could be no variation of the specific heat of a perfect gas with the pressure is demonstrated by Joule's experiment on the throttling of gases (Art. 97) wherein it was shown that in an operation in which no energy in any form was removed from the gas, the temperature remained unchanged, notwithstanding the change in pressure. The heat content of a perfect gas, above absolute zero of temperature, is $c_p T$ (Art. 119). In Joule's experiment, it was shown that the heat content of the gas was unchanged by the operation. Therefore $c_p T_1 = c'_p T_2$ where c'_p is the specific heat at the lower or final pressure,

and

$$T_1 = T_2$$

Hence

$$c_p = c'_p$$

That is to say, the specific heat is independent of the pressure.

Experiments have shown that, in general, the specific heat of gases may be represented by the expression

$$c = a + bt$$

If the equation is multiplied by m, the molecular weight of the gas,

$$mc = ma + mbt$$

But the product mc has been found to have the same numerical value for all simple gases. Hence,

$$mc = A + Bt$$

where A and B are constants for air, oxygen, nitrogen, hydrogen, or any other of the so-called permanent gases.

Langen gives the following:

$$mc_v = 4.77 + .000667t$$

$$mc_p = 6.75 + .000667t$$

where m is the molecular weight,

c_p and c_v the mean specific heats at constant pressure and constant volume, respectively, between 0° F. and t° F.

The quantity mc is called the molecular specific heat. The important fact that this product is constant for all the simpler gases is explained by the theory of heat (see Art. 205). The rise of temperature of a body is due to an augmentation of its molecular velocities. For a lighter gas, there are fewer molecules, m, per unit of volume, and consequently less kinetic energy, or heat need be imparted to a unit volume of the gas to increase the molecular velocity a given amount, than would be the case with a heavier gas. That is to say, if specific heat were defined as the amount of heat necessary to raise a unit volume of a substance 1°, then the specific heats of different gases would vary directly as their molecular weights. However, since the definition of specific heat applies to unit weight of a gas, specific heats will vary inversely as their molecular weights; hence the product of the molecular weight of a gas, multiplied by its specific heat, is the same for all the permanent gases.

Equations showing the variation of the specific heats of CO_2 and H_2O, were also determined by Langen.

For CO_2

$$c_v = .15 + .000066t$$

$$c_p = .195 + .000066t$$

For H_2O

$$c_v = .324 + .000133t$$

$$c_p = .435 + .000133t$$

The above expression for the specific heat of H_2O gas, or steam, is useful where very high temperatures are experienced. The Steam Tables furnish more accurate values for the specific heat at constant pressure within the range of temperature covered by them. The specific heat of steam, and in fact of any gas, is influenced by pressure as well as by temperature when relatively near its point of liquefaction. The value of the specific heat first diminishes with the degree of superheat due to the predominance of the pressure effect, and then increases as the effect of temperature is able to assert itself, thus conforming more nearly in behavior to a perfect gas. The variation of c_p for steam is clearly shown by the Chart in Marks and Davis' Steam Tables.[1]

133. Mixture of Air and Water Vapor.—Atmospheric air always contains water vapor to a greater or less degree. If a cubic foot of atmospheric air be entrapped, and retained in a vessel, the temperature and pressure of the sample will of course be that of the atmospheric air. If the vapor could be separated from the air, and removed from the vessel, the air would then fill the vessel at a lower pressure than the mixture; and likewise, if the air were removed, leaving the vapor, the latter would fill the vessel at a reduced pressure. Let P_t represent the original or barometric pressure; P_a the pressure of the air if the vapor were all removed; and P_s, the pressure of the vapor if the air were all removed. Then

$$P_a + P_s = P_t$$

according to the law of partial pressures.

For a mixture of air and vapor it is not usually practicable to obtain the proportion of the component parts in terms of either volume or weight by direct experiment. An instrument consisting of a wet- and a dry-bulb thermometer, called a psychrometer, is usually employed to indicate the proportion of moisture. The dry-bulb thermometer

[1] Page 97, 1916 edition.

registers what is called the temperature of the air; the wet-bulb, covered with a thin cloth sack, and moistened with water, measures the "wet-bulb temperature" of the air, which in general is lower than the dry-bulb temperature, because of evaporation of some of the moisture. It is from the relation of these two measurements, and reference to suitable tables or formulas, that the proportions of air and vapor in the mixture are finally arrived at.

134. Dew Point.—The water vapor of the mixture is in reality steam, and for it the steam tables are available from which its properties can be found, provided sufficient data are furnished by the observations.

The only thing known at once about the steam of the mixture is its temperature, and that moisture can be evaporated in its presence. The latter fact proves that the steam is in the superheated state. Its condition is therefore represented by some point A on the temperature-entropy diagram of Fig. 79. But only the height of point A is known together with the fact that it lies in the superheat region; nothing is known about its entropy, or pressure, or anything else as yet. If the steam were cooled at constant pressure, as is the case if cooling occurs in the atmospheric air, the path of the change would be along the constant pressure line AB, but not until the temperature is lowered to B, would condensation begin. The temperature at B is called the *dew point*. The dew point can be found by direct experiment. The wet- and dry-bulb instrument, however, does not give it directly. The wet-bulb temperature is above B on AB, at some such point at C. The relation among the various pressures and temperatures

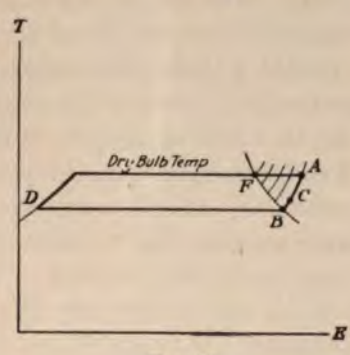

FIG. 79.

involved has been worked out by Carrier,[1] in the formula given below:

$$e = e' - \frac{(P - e')(t - t')}{2755 - 1.28t'}$$

Where t = dry-bulb temperature in degrees F.;
t' = wet-bulb temperature in degrees F.;
e' = vapor pressure corresponding to t' in pounds per sq. in.;
e = vapor pressure corresponding to t_0, the dew point;
P = barometric pressure in pounds per sq. in.

With the dew point known, that is to say the saturated temperature of the steam, the partial pressure, P_s, of the steam, and all the other properties, can be found in the Steam Tables, and the point A can now be definitely located.

135. Relative Humidity.—Referring again to the imaginary cubic foot sample of atmospheric air of Art. 133, if the air could be separated from the steam, the latter would then fill the space at the pressure P_s, the state being represented by the indefinitely located point A, of Fig. 79. Now suppose more steam at the dry-bulb temperature be introduced into the box. The pressure will be raised, as well as the density, by the crowding in of the additional quantity. The operation will be represented by the movement of the state point to the left from A toward F on the constant-temperature line. Higher and higher pressure lines are encountered until F is reached. After this, if the operation continues, and the constant temperature is to be retained, the pressure can no longer rise; in order to prevent its doing so, some of the steam must be condensed to make room for the addition. F is called the saturation point, and represents the maximum weight of steam that can be concentrated in the cubic foot space, as steam. This is true whether the steam exists alone in the vessel, or in mixture with the air.

[1] Trans. A. S. M. E., Vol. 33, p. 1005.

The ratio of the weight of the steam at A to the maximum weight that can be contained at F, at the same temperature, is called the *relative humidity*. The actual weight of steam contained in the cubic foot space, as represented by A, expressed in pounds, or more usually in grains, is called the *absolute humidity*.

With the dew point known, from the Carrier equation of Art. 134, the weight of the cubic foot of steam at A, observed conditions, can be found from the Steam Tables; and also the weight of unit volume at saturated condition F. The ratio of the weights gives the *relative humidity* at state A.

136. Determination of Weight of Steam and Air in a Cubic Foot of the Mixture.—For illustration, assume that the dry-bulb temperature is observed to be 80°; the wet-bulb temperature, 70°; and the barometric pressure 14.45 pounds per sq. in. What are the weights of the steam and pure air respectively in a cubic foot of the mixture?

In the equation of Art. 134,

$$e = e' - \frac{(P-e')(t-t')}{2755-1.28t'}$$

$$t = 80°$$

$$t' = 70°$$

$$e' = .3626$$

$$P = 14.45$$

Whence

$$e = .310 \text{ pound per sq. in.}$$

Referring to the Steam Tables, the dew-point temperature, corresponding to .310 pound per sq. in., is found to be 65.4°. The actual steam under consideration is therefore at the pressure of .310 pound per sq. in. and 14.6 degrees superheat. Inasmuch as the Steam Tables do not give the properties of superheated steam at pressures lower than one pound, how can the weight of the cubic foot of steam

at observed conditions be determined? The simplest method is to make use of a modification of the formula employed in the computation of the specific volumes of superheated steam for the Steam Tables.* At low pressures, the equation may be written in the simple form below, without appreciable error,

$$PV = .5962\ T$$

or

$$V = \frac{.5962\ T}{P}$$

where P = pressure in pounds per sq. in.
T = absolute temperature,

and

V = specific volume in cu. ft.

The density, or weight per cubic foot, is the reciprocal of specific volume, and may be expressed thus:

$$\text{density} = 1.68\frac{P}{T}.$$

Applying this equation to the example, the weight of the cubic foot of steam at .310 pound and 80° is found to be .000964 pound or 6.75 grains.

The weight of a cubic foot of saturated steam at 80°, the dry-bulb temperature, is .001570. The relative humidity is therefore .000964 ÷ .001570 = .61 or 61 per cent.

The partial pressure of the air is now obtainable:

$P_a = P_t - P_s = 14.45 - .310 = 14.140$ pounds per sq. in.

From the characteristic equation, the weight of the air in the cubic foot can be determined.

$$W_a = \frac{PV}{RT} = \frac{14.140 \times 144 \times 1}{53.34 \times 540} = .070661.$$

The weight of the steam, $W_s = .000964$.

The weight of the mixture per cu. ft. is

$$W_a + W_s = .070661 + .000964 = .071625.$$

* $PV = 0.5962\ T - P(1 + 0.0014\ P)\left(\frac{150{,}300{,}000}{T^3} - 0.0833\right)$. Marks and Davis Steam Tables, 1916 Edition, p. 98.

The proportions of the constituents by weight are:

Air.....	98.66 per cent
Steam.....	1.34 per cent

137. Specific Heat of Gas and Vapor Mixture.—The specific heat of the mixture is determined from the weights and specific heats of the individual constituents. Continuing with the example of air and vapor mixture, the specific heat of the superheated steam at constant pressure is about .45, and that of air .241.

Specific heat at constant pressure, of the mixture

$$=\frac{(.070661\times.241)+(.000964\times.45)}{.071625}=.244$$

138. Effect of Compression upon Humidity—Isothermal Compression.—If a mixture of air and vapor is compressed, the relative humidity will be changed. Let the mixture whose composition has been worked out under atmospheric conditions, be assumed to be compressed isothermally to 100 pounds per sq. in. abs. The final temperature will then be 80°. The tendency will be for the pressures of the two substances to rise in proportion to the decrease in volume In Fig. 80 the state point of the air, starting from *A* proceeds to the left, crossing progressively higher pressure lines until it arrives at the pressure of compression at *H*.

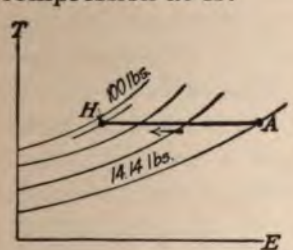

FIG. 80.

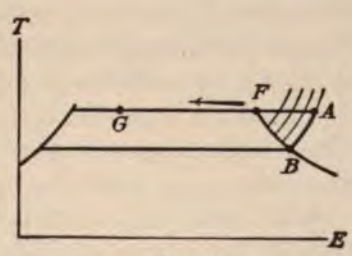

FIG. 81.

In Fig. 81, the state point of the steam, starting from *A* proceeds to the left into the region of higher pressure

until F is reached, after which the pressure remains constant, and some of the steam is condensed, the final state being represented by some point G. The amount of condensation can be computed.

At the end of compression, the partial pressure of the steam is $P_s = .505$ pound per sq. in. corresponding to 80°. The partial pressure of the air is

$$100.00 - .505 = 99.495$$

The weight of the air, in the original cubic foot is .070661 pound and that of the steam .000964 (Art. 136). The volume occupied by the compressed air is:

$$V = \frac{W_a R T_2}{P_2} = \frac{.070661 \times 53.34 \times 540}{99.405 \times 144} = .142 \text{ cu. ft.}$$

The steam occupies the same volume. If it were dry and saturated, a cubic foot would contain .001570 pound and .142 cu. ft. would contain .000223 pound. Since there is present .000964 pound, it is evident that only a portion of the entire amount can exist as steam or vapor; the remainder is condensed. The proportion of vapor or the quality is:

$$q = .000223 \div .000964 = .23 \text{ or } 23 \text{ per cent}$$

The weight of steam condensed, out of each cubic foot of free air as a result of compression and simultaneous cooling is $.000964 \times .77 = .000742$ pound. The same effect would be obtained by any kind of a compression followed by cooling to the original temperature. The water resulting from condensation, can be drained from the lines or receiver, leaving the air more nearly free of moisture, which is usually an advantage no matter to what service the compressed air is put.

139. Effect of Compression upon Humidity—Adiabatic Compression.—In adiabatic compression, the temperature

of the mixture will rise, and the pressure of both constituents will increase indefinitely, the steam becoming more highly superheated. With the relatively small proportion of moisture in atmospheric air and further due to the fact that the steam becomes more highly superheated as a result of the operation, it will be sufficiently accurate to consider the mixture as following the laws of a perfect gas.

Referring to the specific case that has already served as an example, the density of the mixture was found to be .071625 (Art. 136).

$$R = \frac{PV}{WT} = \frac{14.45 \times 144 \times 1}{.071625 \times 540} = 53.81$$

The specific heat of the mixture at constant pressure is approximately

$$c_p = .244 \text{ (Art. 137)}$$

$$c_v = c_p - \frac{R}{J} = .244 - \frac{53.81}{778} = .175$$

$$k = c_p \div c_v = 1.40$$

For adiabatic compression, the final temperature of the mixture can now be computed, and is found to be 944° absolute or 484° F., assuming the terminal pressure to be 100 pounds abs. The original 1 cu. ft. has been reduced in size to .251 cu. ft. The weight of the steam in the mixture is .000964 pound (Art. 136). Hence the specific volume, or volume per pound of the steam after compression, is 260 cu. ft. Knowing the specific volume and the temperature of the steam, reference to Diagram II in the back of the Steam Tables, shows the pressure of the steam to be 2.17 pounds per sq. in., and the superheat about 356°.

It is evident that the same method may be applied to compression along any polytropic line, or to an expansion instead of a compression.

Problems

1. The analysis of a gas shows the following composition by volume:

$CO = 28.12$ per cent
$CH_4 = 1.50$ per cent
$H_2 = 3.76$ per cent
$CO_2 = 5.41$ per cent
$N_2 = 61.21$ per cent

Determine the composition by weight, the molecular weight, and the value of R for the gas.

2. The analysis of a furnace gas, by volume, shows that it is composed of,

$CO_2 = 12.51$ per cent
$O_2 = 6.12$ per cent
$H_2O = 2.80$ per cent
$N_2 = 78.57$ per cent

Determine the composition by weight, the molecular weight, the value of R, and the specific heat at constant pressure of the mixture if the temperature is 550° F.

3. The dry-bulb temperature in a room is 70°, the wet-bulb temperature 56°, and the barometric pressure 29.15. Determine the following:

a. Weight of the two constituents, steam and air, and the total of the mixture, per cu. ft.

b. Relative humidity in per cent, and the absolute humidity in grains per cu. ft.

c. If this air mixture is heated to a temperature of 150°, what will be its relative humidity? If used for drying purposes, how many grains per pound of air supplied could possibly be absorbed by it?

4. A fan is handling air at a temperature of 78° and a barometric pressure of 29.40 in. The wet-bulb temperature is 65°, what error, in per cent, will be made in computing density, if it be assumed that the substance is pure dry air (the humidity factor being entirely neglected), instead of a mixture of air and steam?

5. Which is the heavier; a cubic foot of dry air or a cubic foot of saturated air at the same dry-bulb temperature?

6. A compressor handles 400 cu. ft. of free air per minute, compressing it from atmospheric conditions to 100 pounds gage pressure. The barometric pressure is 29.60 in. Hg, 75° dry-bulb temperature, and 60° wet-bulb temperature. The water-jacketing of the compressor cylinder together with cooling after compression reduces the temperature to the original atmospheric temperature, viz., 75°. What weight of water will be condensed per minute due to the compression and cooling in the manner described?

CHAPTER XIII

THE AIR-HEAT ENGINE

140. The Internal-combustion Engine is an Air Engine. —In a compressed-air system the delivery from the compressor is usually cooled by radiation back to the atmospheric temperature of the air. The utilizer or air engine may receive the compressed air at this lower temperature level, and mechanical energy will be derived from it. The work thus obtained will be less than that put in by the compressor, overlooking all frictional losses. because of the loss of heat by radiation.

Usually the compressed air is preheated before it is received by the air engine, thereby utilizing the air as a working substance or a medium for the transformation of heat into work. The air engine may then deliver a larger quantity of power than was used to run the compressor.

The internal-combustion engine is essentially a heat engine that combines in one unit the three elements of the compressed-air system alluded to above, viz., the compressor, the heater, and the heat utilizer. The work of expansion is performed upon the same piston that does the work of compressing the charge. The heating is done by the combustion of a fuel mixed with the working air itself. It is the latter feature that has made this type of air engine a success. All attempts to build and operate air engines as heat engines for primary sources of power, by introducing the heat from an external source through separating walls have proved failures. The rate of heat transfer per unit of area is too slow for large capacities; and the temperatures desirable for good efficiency are apt to be

destructively high for the material of the heat transmission surfaces. The external heater arrangement is satisfactory as a preheater in a compressed-air system where the heating effect is rather incidental; but it is impracticable in its application to air engines considered as prime movers.

The working substance of the internal combustion engine is, of course, not pure air. For accurate analysis, the substance is to be treated as a mixture. On the other hand, since the predominating element of the mixture under all conditions is nitrogen, which is also the case with air, the properties of the mixture will never diverge very greatly from those of air. For preliminary or tentative calculations it is therefore sufficiently accurate to consider the working substance as air.

141. The Otto Cycle.—Most internal-combustion engines operate on the Otto cycle. The ordinary four-cycle, or more correctly speaking, four-stroke-cycle engine, draws in its charge of air, with which the fuel is usually mixed, on the first stroke; compresses the charge on the second stroke, after which ignition and development of the heat takes place; receives the work of expansion of the gas on the third stroke; and ejects the residue of the charge, after release, on the fourth stroke, thus completing the entire cycle in four strokes or operations of the piston.

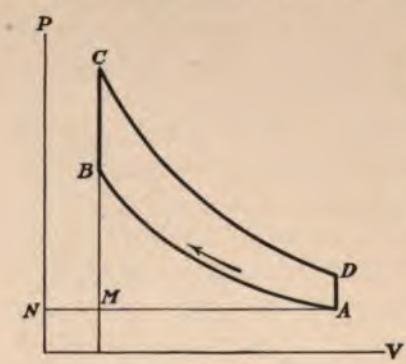

FIG. 82.

The ideal indicator diagram of a four-stroke-cycle Otto engine is illustrated in Fig. 82. *MA* represents the introduction of the charge; *AB* is adiabatic compression; *BC* is instantaneous combustion of the fuel and heating of the charge at constant volume; *CD* is adiabatic expansion of the hot gas; *DA* is an instantaneous drop of pressure fol-

lowing release; and AM is the ejection of the remainder of the charge.

The actual indicator diagram departs from the ideal chiefly because the cylinder must be water-jacketed to keep the temperature of rubbing surfaces sufficiently low to allow effective lubrication. The compression and expansion lines are not adiabatics. The burning of the fuel, although extremely rapid, cannot be instantaneous, neither can the theoretical temperature of combustion, represented by C, be reached, because of the simultaneous cooling of the charge.

The ideal indicator diagram of a two-stroke-cycle Otto engine is essentially like that of the four-stroke cycle. The only difference is the absence of the intake and discharge operations, MA and AM, of Fig. 82, in the two-stroke cycle. Since those operations just balance each other in energy terms, they have no significance in the theoretical diagram.

142. *PV* and *TE* Diagrams of the Otto Cycle. — Although Fig. 82 is a representation of the changes of pressure and volume that would take place *within the cylinder* of an ideal Otto engine, it is not a correct representation of all the changes in pressure and volume of the working substance which are pictured more nearly correctly in Fig. 83. The temperature-entropy changes are shown in Fig. 84.

Fig. 83 is just like Fig. 82, following the cycle around to D, the point of release. At the state D the cylinder may be considered as a reservoir of compressed air or gas, not in communication with any source. When the exhaust valve opens, there is an expansion like that described and discussed in the example of Arts. 121, 122 and 123. There is a growth of volume from D, not an actual removal of heat at constant volume. Work is done by the substance in displacing the atmosphere to make room for itself. The energy for the work done must come from the intrinsic energy, originally possessed by the charge at D. Conse-

quently, the average temperature of the charge immediately following release is represented by F, lower than H, which is on the isothermal DH through D (Figs. 83 and 84) and on the atmospheric pressure line AF.

Let it be supposed that the volume of the exhaust piping from the cylinder to the place of escape into the atmosphere is large in comparison to that of the cylinder. Upon release at D (Fig. 83) the charge, which has been confined completely within the cylinder, now occupies both the cylinder volume and a part of the exhaust piping. Its state at this moment is represented by F, Figs. 83 and 84.

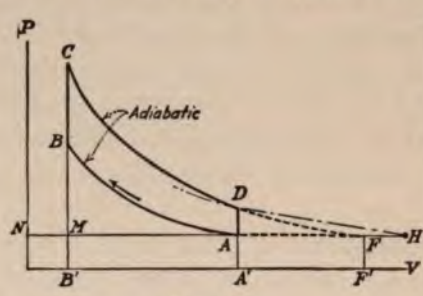

FIG. 83.

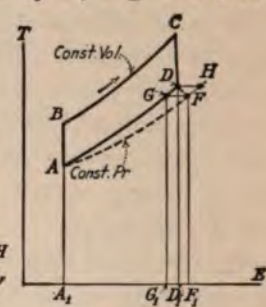

FIG. 84.

If a sufficient length of time be allowed, the charge will cool by loss of heat through the walls of the exhaust piping at constant atmospheric pressure to the point A, which is the state of the charge at the beginning of the cycle. The heat lost during this cooling, which is the heat rejection operation of the cycle, is represented by the expression

$$\overline{W}c_p(T_F - T_A)$$

In the ideal Otto Cycle, the heat rejection operation is usually represented as a cooling at constant volume from D to A (Figs. 83 and 84). If it can be shown that

$$\overline{W}c_v(T_D - T_A) = \overline{W}c_p(T_F - T_A)$$

then the representation of the heat operation of the cycle as a constant-volume cooling may be accepted and used because of its simplicity.

When the charge is released at D (Fig. 83), work is done by it in displacing the atmospheric air, to the extent of

$$\text{area } A'AFF' = P_A(V_F - V_A)$$

T_F is the average temperature of the charge immediately after release. Let $\overline{W}$ = the weight of the charge.

Then, loss of intrinsic energy $= \overline{W}c_v(T_D - T_F)$.

But

$$Jc_v\overline{W}(T_D - T_F) = P_A(V_F - V_A).$$

In Fig. 84 take the point G on the constant-volume line through D, at the same temperature as F.

Then

$$\overline{W}c_v(T_D - T_F) = \overline{W}c_v(T_D - T_G)$$

Hence

$$\frac{1}{J}P_A(V_F - V_A) = \overline{W}c_v(T_D - T_G) = \text{area } D_1\,DGG_1$$

Also

$$\overline{W}c_p(T_F - T_A) = \overline{W}c_v(T_F - T_A) + \frac{1}{J}P_A(V_F - V_A) \text{ (Art. 101)}$$

$$= \overline{W}c_v(T_G - T_A) + \frac{1}{J}P_A(V_F - V_A)$$

Therefore

$$F_1FAA_1 = G_1GAA_1 + D_1DGG_1 \text{ (Fig. 84)}$$

Or

$$\overline{W}c_p(T_F - T_A) = \overline{W}c_v(T_D - T_A)$$

The heat that must be given up by the charge following "free" expansion at D, in order that it may be cooled to the original state A, at constant pressure, is therefore equal to the heat that would be withdrawn in cooling the charge at constant volume to the original state of the cycle.

In an actual engine employing the Otto cycle, following release, when a part of the charge escapes from the cylinder by its own elasticity, the piston promptly pushes out the remainder of the charge into the atmosphere. Without waiting for the identical charge to be cooled to its original state, the piston takes in another charge from the atmospheric reservoir, while the first is cooled at leisure at constant pressure by commingling with the outside air.

143. Heat Quantities of the Otto Cycle.—The Otto Cycle is again shown in Fig. 85 and 86, where the heat

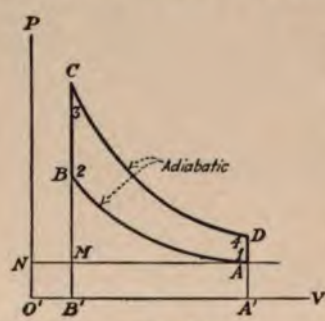

FIG. 85.

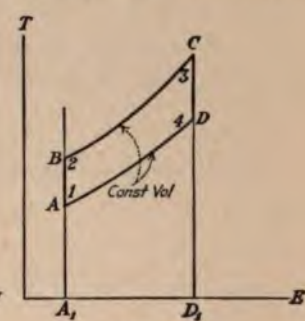

FIG. 86.

removal is represented as one of constant volume, which has just been shown to be the equivalent of the actual operation.

Consider the energy equation for gases,

$$H = (S_2 - S_1) + \frac{W}{J}$$

Where H = the quantity of heat supplied to the working substance. $(S_2 - S_1)$ = the increase in intrinsic energy and $\frac{W}{J}$ = the heat equivalent of work done by the gas. Let this equation be applied to each of the four operations of the Otto cycle. It will be convenient to place the appropriate expressions in tabular form, and to indicate the direction of the operation by words at the top of the column

rather than by the sign of the arithmetic answer. The symbol $\overline{W}$ is used to designate the weight of the charge of working substance.

Operation.	Heat Quantity—H.			Intrinsic Energy Change (S_2-S_1)	Heat Equivalent of Work—$\frac{W}{J}$.		
	Supplied.	Rejected.	Area.		*By* the Charge.	*On* the Charge.	Work Area.
AB	0	0	.....	$\overline{W}c_v[T_2-T_1]$	0	$\frac{P_2V_2-P_1V_1}{J(k-1)}$	$A'ABB'$
BC	$\overline{W}c_v[T_3-T_2]$	0	A_1BCD_1	$\overline{W}c_v[T_3-T_2]$	0	0	.....
CD	0	0	.....	$\overline{W}c_v[T_4-T_3]$	$\frac{P_3V_3-P_4V_4}{J(k-1)}$	0	$B'CDA'$
DA	0	$\overline{W}c_v[T_4-T_1]$	D_1DAA_1	$\overline{W}c_v[T_1-T_4]$	0	0	.....
Total	$\overline{W}c_v[T_3-T_2]$	$\overline{W}c_v[T_4-T_1]$	$ABCD$ (Fig. 86)	0	$\frac{P_3V_3-P_4V_4}{J(k-1)}$	$\frac{P_2V_2-P_1V_1}{J(k-1)}$	$ABCD+J$ (Fig. 85)

The excess of heat supplied from the source over the heat rejected to the refrigerator must equal the excess of work done *by* the gas over work done *on* the gas, because there is no net change in the intrinsic energy of the gas after completing the cycle.

$$\overline{W}c_v(T_3-T_2)-\overline{W}c_v(T_4-T_1)=\frac{P_3V_3-P_4V_4}{J(k-1)}-\frac{P_2V_2-P_1V_1}{J(k-1)}$$

But

$$\frac{P_3V_3-P_4V_4}{J(k-1)}=\overline{W}c_v(T_3-T_4)\text{ (Art. 103)}$$

and

$$\frac{P_2V_2-P_1V_1}{J(k-1)}=\overline{W}c_v(T_2-T_1)$$

Therefore

$$\overline{W}c_v(T_3-T_2)-\overline{W}c_v(T_4-T_1)=\overline{W}c_v(T_3-T_4)-\overline{W}c_v(T_2-T_1)$$

or

$$(T_3-T_2)-(T_4-T_1)=(T_3-T_4)-(T_2-T_1)$$

which is obviously an identity, and verifies what is already known, viz., that the excess of heat supplied ever heat rejected is equal to the excess of work done by the gas over work done on the gas.

144. Efficiency of the Otto Cycle.—

$$\text{Efficiency} = \frac{\text{heat supplied} - \text{heat rejected}}{\text{heat supplied}}$$

$$= \frac{\overline{W}c_v(T_3 - T_2) - \overline{W}c_v(T_4 - T_1)}{\overline{W}c_v(T_3 - T_2)}$$

$$= 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

The efficiency equation may be manipulated to show the very important fact that the efficiency is dependent upon the pressure of compression.

In Fig. 85 AB and CD are adiabatics.

Hence

$$P_2V_2^k = P_1V_1^k \quad \text{and} \quad P_3V_3^k = P_4V_4^k$$

Or

$$P_2V_2V_2^{k-1} = P_1V_1V_1^{k-1} \quad \text{and} \quad P_3V_3V_3^{k-1} = P_4V_4V_4^{k-1}$$

But

$$V_3 = V_2 \quad \text{and} \quad V_4 = V_1; \text{ also } P_2V_2 = RT_2, \text{ etc.}$$

Dividing one equation by the other and substituting:

$$\frac{T_3}{T_2} = \frac{T_4}{T_1}$$

$$T_3T_1 = T_4T_2$$

Subtract the quantity T_1T_2 from both sides of the equation

$$T_3T_1 - T_1T_2 = T_4T_2 - T_1T_2$$

$$T_1(T_3 - T_2) = T_2(T_4 - T_1)$$

$$\frac{T_4 - T_1}{T_3 - T_2} = \frac{T_1}{T_2}$$

Hence,

$$\text{efficiency} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1}{T_2}$$

Again, by the adiabatic relation between A and B,

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}. \quad \text{(Art. 90)}$$

Therefore,

$$\text{efficiency} = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}}$$

From this expression it is clear that the efficiency of an ideal Otto, or similar cycle, increases as P_2, the pressure of compression, increases.

The same important fact can be demonstrated graphically with the help of a temperature-entropy diagram of an ideal Otto cycle. In Fig. 87 let $ABCD$ represent the cycle of an engine of a given clearance volume, which establishes the compression pressure P_2 at B. Now imagine the clearance to be reduced by some change in the engine, leaving the total initial volume at A unchanged. The change in the engine might be accomplished by lengthening the stroke, and adjusting the position of the cylinder so that the extreme point of travel of the piston with reference to the cylinder should be the same as before the change, thus reducing the clearance volume, but insuring that the same total volume, and therefore the same mass of charge, will be acted upon per stroke. After the change the initial state A, is the same as before, and the compression line is coincidental with AB but proceeds to the higher pressure at F. The charge is heated along the constant volume line FG, to G. Let it be assumed that the amount of heat supplied is the same after the change as before. Then the area A_1FGM_1 = area A_1BCD_1. From G, the charge expands adiabatically to M, which necessarily lies

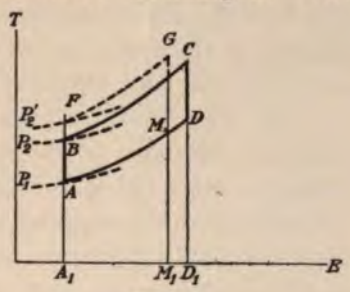

FIG. 87.

on the line DA, since the charge is finally restored to the original state A by constant volume cooling.

Let H_1 = the amount of heat supplied in both cases. Before the change,

$$\text{efficiency} = \frac{A_1BCD_1 - D_1DAA_1}{A_1BCD_1} = \frac{H_1 - D_1DAA_1}{H_1}$$

After the change,

$$\text{efficiency} = \frac{A_1FGM_1 - M_1MAA_1}{A_1FGM_1} = \frac{H_1 - M_1MAA_1}{H_1}$$

But M_1MAA_1 is less than D_1DAA_1. Hence the efficiency is higher as a result of the higher compression pressure.

A greater portion of the supplied heat in the cycle with this increased compression pressure is distributed above the line of cooling through A, which separates the supplied heat into an available and an unavailable area. The effect of the higher compression is to elevate the region of activity of the cycle, on the temperature-entropy plane, and so reduce the proportion of rejected to supplied heat.

145. The Diesel Cycle.—In 1895, Rudolph Diesel, a German, brought out an internal combustion engine that operates upon what is now known as the Diesel cycle. The Diesel engine is particularly adapted to the successful use of fuel oils of almost any grade, and on account of its excellent efficiencies, it is steadily gaining in use.

The cycle is represented on the PV and TE planes in Figs. 88 and 89. Diesel engines are built to operate upon either the four-stroke or two-stroke-cycle, but in either, the ideal cycle consists of adiabatic compression AB; constant pressure heating BC; adiabatic expansion CD; and constant volume cooling DA.

In addition to the difference between the Diesel and Otto cycles in the nature of the second operation, the Diesel engine is characterized by the very high compression pressures employed. In the Diesel engine, the fuel that is to

supply the heat is not mixed with the air upon its introduction into the cylinder. The compression operation *AB*, is performed upon a charge of air only. As a result there is no danger of preignition during compression. The pressure of compression is always limited when the fuel is previously mixed with the air. In the Diesel engine there is no limit other than the mechanical limitations of pressure and temperature.

At *B* the oil is injected, and burning takes place until the supply of fuel is cut off. The ideal Diesel cycle is produced when the rate of injection is so balanced with the rate at which the gas is doing work that a constant

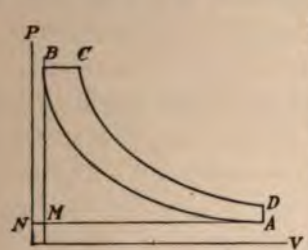

FIG. 88.

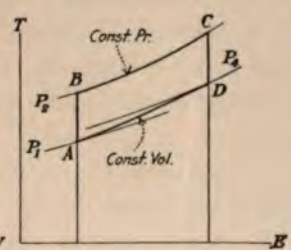

FIG. 89.

pressure is maintained. If the rate of injection is less than this balancing amount, the pressure will fall during the operation *BC*; if it is greater, the pressure willl rise.

The heat quantities and efficiency of the Diesel cycle can be worked out in a manner similar to that used for the Otto cycle. The efficiency of the Diesel cycle is higher than that of the Otto, because of the higher compression pressures that can be employed.

146. The Brayton Engine.—Practically every internal combustion engine at the present time operates upon either the Diesel or the Otto cycle, or possibly upon a cycle that is a combination of the two. The Diesel represents the latest cycle to be developed, while the Otto is the most widely used. The first Otto engine was built in 1876, and was known as the Otto "silent" engine. The word

" silent " is a relative term, and was used presumably to emphasize the relief at not having to listen to the Otto and Langen free-piston engine, which was the immediate forerunner of the Otto engine. The Otto and Langen engine operated upon the Otto cycle. As a matter of fact, the Otto cycle was not invented by Otto, but was suggested and discussed about 1856 by Beau de Rochas, and is sometimes known by his name.

A few years before the Otto " silent " engine was developed, an American named Brayton developed an engine that met with some success commercially until it was displaced by the Otto engine, which proved superior to it. The cycle of the Brayton engine is sometimes called the Joule cycle, because Joule had suggested it before the engine was developed by Brayton.

Fig. 90 gives some idea of the mechanical elements of the engine.

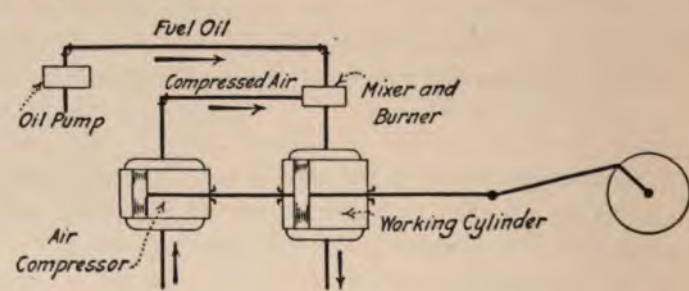

FIG 90.

The air is compressed in the small cylinder. The compressed air and fuel oil are mixed and burned, and then admitted to the working cylinder, where work is done both by the pushing in of the charge, and the further expansion of the charge after cut off, after which the spent gases are exhausted into the atmosphere.

The cycle of the Brayton engine is represented on the PV and TE planes in Figs. 91 and 92. $MABN$ is the work done in compressing the air; $NCDM$ is the work done by

the charge; and *ABCD* is the net work delivered by the engine.

147. The Lenoir Cycle.—The first commercially successful internal-combustion engine was built in 1830 by Lenoir, a Frenchman. The Lenoir cycle is pictured in

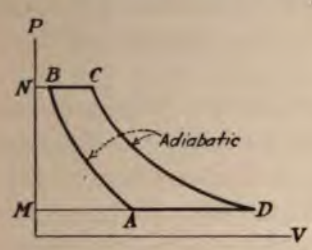

FIG. 91.

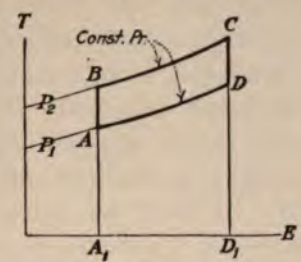

FIG. 92.

Figs. 93 and 94. The engine drew in a mixture of fuel and air along *MA*, Fig. 93. At about half stroke the charge was ignited, producing, theoretically, constant-volume heating, *AB*. Adiabatic expansion then occurred, along *BC*, followed by the equivalent of constant volume cooling

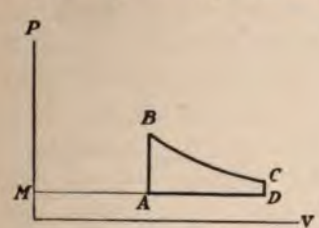

FIG. 93.

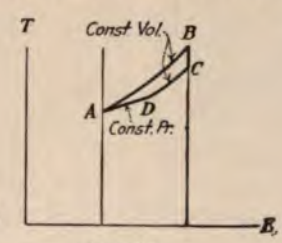

FIG. 94.

after release at *C*, and then constant-pressure cooling to *A*. With no compression of the charge, it is obvious that the efficiency cannot be very high. The Lenoir engine could not compete in economy with the Otto and Langen free-piston engine, developed in 1867, notwithstanding its superiority over the latter in mechanical operation. The importance that attaches to the Lenoir engine is the fact that it stands as the first successful internal-combustion

engine emerging from a long period of study and experiment devoted to the air-heat engine by scientists and inventors.

148. The Stirling "Hot-air" Engine.—Forty-four years before the appearance of the Lenoir internal-combustion-air engine, the first external-combustion-air engine, or as it is commonly called, hot-air engine, was built by Stirling, an English preacher.

The engine consisted primarily of two cylinders A and B, Fig. 95. In the cylinder A the working charge received and rejected heat; while in the cylinder B the power was developed. A furnace, F, is represented as the source of heat which is applied at the bottom of cylinder A. C is a displacer piston, whose function is to move the charge from one end of the cylinder A to the other, at the proper time. The air, upon being displaced from one end of A, passed through the checker-work heat-regenerator R, to the other end. In the upper end of A was a cooling coil D. All the necessary elements of a heat engine are thus accounted for. F is the source of heat energy; D is the refrigerator into which unavailable and waste heat was rejected; B is the heat utilizer; and the working substance is air, of which a charge is permanently confined within the engine.

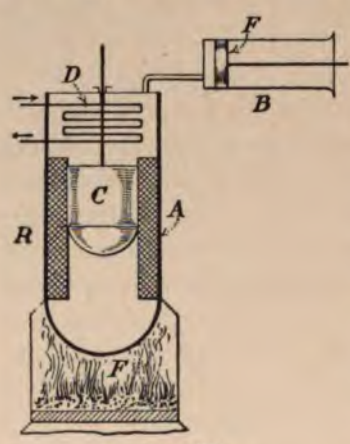

FIG. 95.

Assume, to begin with, that the displacer piston C is at the top of its stroke, as shown in the figure, Nearly all the air in cylinder A is now at the bottom, in contact with the heating surface. The working piston F is at the beginning of its stroke, at the left. With the heating of the

charge, the air expands, pushing out the piston F, and the expansion, which is supposed to be isothermal, continues until F reaches the end of its stroke. At that moment C descends, displacing the air below it to the space above. The heated air, in passing through R, parts with some heat, which is stored up to await the opportunity to return to the working substance. The thermodynamic operation with respect to the air is one of constant-volume cooling. It is to be understood that during the movement of C the working piston F is dwelling at its extreme position, and that the quantity of air in A is much larger than in B, so that it is approximately true to say that the heating operations which take place in A apply to the entire charge of air.

The air, which is now mostly confined in the upper part of A, is subjected to the action of the refrigerator and begins to contract in volume. The working piston is moved back to the left, and isothermal compression is supposed to take place. When F reaches the end of its stroke, C ascends and displaces the air above it, to the bottom of A. The temperature of the air is relatively low as it enters the passage R, and it now picks up the heat that it stored on its upward passage. This is the principle of the heat regenerator, and the Stirling engine was the first apparatus to make use of that principle, which, although never extensively applied in heat-engine work, has proved to be of so much value in furnace operation in other lines of Engineering.

149. The Cycle of the Stirling Engine.—The ideal cycle of the Stirling engine consists of two isothermals and two constant-volume lines, as shown in Figs. 96 and 97. The PV diagram pictures the operations in the working cylinder, while the TE diagram shows the thermal operation in cylinder A. (Fig. 95). AB is the working isothermal; BC is the constant-volume cooling resulting from the air giving up its heat to the regenerator; CD is the isothermal compression; and DA is the constant-volume heating.

The total heat supplied is represented by the area $D_1DAA_1+A_1ABB_1$ (Fig. 97); the heat rejected by area $B_1BCC_1+C_1CDD_1$, and the heat equivalent of the work done, by the difference, viz., $ABCD$. However, the heat quantity B_1BCC_1 is not rejected to the refrigerator, but is temporarily stored up in the regenerator; nor does that

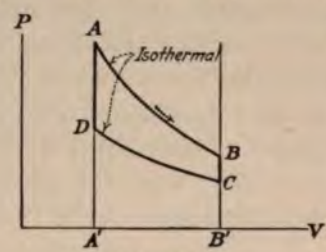

FIG. 96.

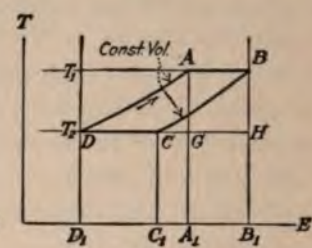

FIG. 97.

portion of the heat supplied, D_1DAA_1 come from the heat source. Ideally, these heat quantities are equal.

$$B_1BCC_1 = D_1DAA_1$$

The length DC = the length AB (Fig. 97). Hence

$$DC = GH \text{ and area } D_1DCC = A_1GHB_1$$

The efficiency of the ideal cycle then is

$$\frac{A_1ABB_1 - A_1GHB_1}{A_1ABB_1} = \frac{T_1 - T_2}{T_1}$$

In the next chapter it will be shown that the maximum possible efficiency for any heat engine is $(T_1 - T_2) \div T_1$. However, in spite of the excellent efficiency of its ideal cycle, the Stirling engine was not a success. The actual timing of the various operations of the engine was far from ideal, and the temperatures that had to be used to force the capacity up to a reasonable point were too severe for the heating surfaces.

150. The Ericsson Hot-air Engine.—Another type of hot-air engine was developed by John Ericsson, a Swede by

birth, who left his native country while still a young man to practice as an engineer in England and later became a naturalized American. He was the inventor and builder of the *Monitor* of Civil War fame. It was while he was in England that he undertook to equip a vessel of 2200 tons for the United States Navy, with his hot-air engines. There were to be four engines each having a single acting working cylinder 14 ft. in diameter, and 6-ft. stroke. The engines ran at 9 R.P.M. and developed 300 H.P. each. They were unsuccessful for the same reasons that caused the failure of the Stirling engine. The heating surface deteriorated under the high temperatures to which they were subjected. In any external-combustion engine, large quantities of heat must be transferred from one substance to another through a separating wall. This is accomplished without difficulty in a steam-boiler plant, but the receiving substance is water whose relatively low temperature protects the transmitting material from undue heating. In the hot-air engine much higher temperatures of the working substance were necessary under normal operating conditions, and at the same time the possibility of accidental overheating was much greater.

151. Cycle of the Ericsson Engine.—The cycle of the Ericsson engine is shown on both the PV and TE planes, Figs. 98 and 99. It is composed of two isothermals and two

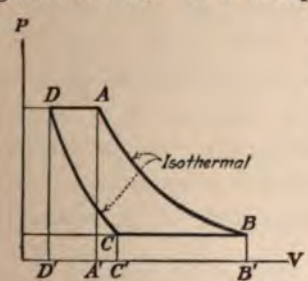

FIG. 98.

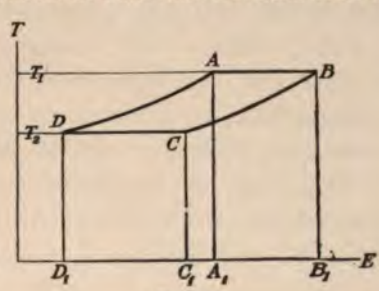

FIG. 99.

constant-pressure lines. The heat A_1ABB_1 (Fig. 99) is supplied along AB and the work $A'ABB'$ (Fig. 98) is done

on the piston. Constant pressure cooling takes place along BC, as the air is depositing the heat B_1BCC_1 in the regenerator; the work $B'BCC'$ is done by the piston upon the charge. The air is then compressed isothermally, having the further work $C'CDD'$ done upon it, while the heat C_1CDD_1 is rejected. It then passes through the regenerator again, picking up the heat D_1DAA_1 ($=B_1BCC_1$, which it had previously deposited) and doing the work $D'DAA'$ by its growth in volume.

It is apparent that the expression for efficiency of the cycle is the same as that derived for the Stirling engine.

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1}$$

The hot-air engine as a heat motor is now all but obsolete. For a time, even lately, it had a use in small capacities as a motor for farm work, chiefly for operating well pumps. But even in that restricted field it has been displaced by the windmill, and later by the more powerful and dependable gasoline or oil engine. The chief interest that the hot-air engine presents is the recognition of the place which it has had in the development of the science of thermodynamics. Both types of hot-air engine were brought out at a time contemporaneous with the developments of Carnot in working out the Second Law of Thermodynamics, and the experiments of Joule in establishing the First. Men were thinking deeply and carefully at that time, and whatever may be the shortcomings of the hot-air engine as a result of the limitations of working mechanisms, thermal conductivity, and refractoriness of material, it cannot be denied that the conception of these cycles reveals a knowledge on the part of the inventors that is truly wonderful.

Problems

1. Consider a compressed-air system such as is represented by Fig. 40, and assume the same pressures and temperature as indicated thereon. Let a heater be installed to heat the compressed air from 70° to 500° at the en-

trance to the air engine. How much more work, per pound of air, will be done by the air engine equipped with the heater than by an engine not equipped with the heater, taking its supply at 70°, assuming adiabatic expansion to the back pressure in both cases? How much more heat is rejected in the exhaust? What is the efficiency of utilization of the heat supplied in the heater?

2. Consider an Otto-cycle gas engine in which natural gas is used as the fuel. The clearance is 28 per cent. Of the charge drawn in on the suction stroke the proportions are 12 cu. ft. of air to 1 cu. ft. of gas. The pressure of the charge in the cylinder, at the end of the suction stroke, is 14 pounds per sq. in. abs., and the temperature is 120°. The heating value of the gas is 900 B.T.U. per cu. ft. at 14.7 pounds pressure and 60° F. Assuming the charge to have the properties of air, and considering the ideal cycle of the Otto engine (Fig. 85):

a. Calculate the heat and work quantities for each of the operations of the cycle, and insert the numerical values (for one pound weight) in a table like that of Art. 143.

b. What is the efficiency of the cycle?

c. If the clearance could be reduced to 20 per cent, thus increasing the compression pressure, calculate the efficiency and compare with (*b*).

3. In a Diesel-cycle engine, let the charge of air be compressed from the suction pressure of 14 pounds per sq. in. abs., and 130°, to 500 pounds abs. At a given load, the engine uses 20 pounds of air per pound of oil. The heating value of the oil is 19,000 B.T.U. per pound. Calculate the heat and work quantities of the various operations and the efficiency for the ideal cycle, assuming the charge to behave as though it were air.

CHAPTER XIV

THE ENERGY LAWS OF THERMODYNAMICS

152. The Definition of Thermodynamics.—The definition of Thermodynamics given in Art. 1 is repeated here; " Thermodynamics is that branch of the theory of heat that treats of the relations between heat and mechanical work. especially as acting in a heat engine." Although energy appears in many forms, the above definition limits consideration to but two of these, viz., heat and mechanical work.

Two fundamental statements commonly known as the *First and Second Laws of Thermodynamics*, can be made concerning the relations between heat and mechanical work.

153. The First Law of Thermodynamics.—The First Law of Thermodynamics states that: *Energy in the form of heat may be converted into mechanical work, and energy in the form of mechanical work may be converted into heat.* A definite relation exists between the value of a heat unit and a work unit. In the English system of units this definite relation is 1 to 778; that is

$$1 \text{ B.T.U.} = 778 \text{ foot-pounds}$$

A foot-pound is the energy required to raise a weight of one pound one foot. A British Thermal Unit (B.T.U.) is equal to $\frac{1}{180}$ of the heat required to raise the temperature of one pound of water from 32° F. to 212° F.

The relation between the energy values of a B.T.U. and a foot-pound rests upon experimental evidence. The

experiment was first performed by Joule in 1843, who used a device whereby a measured amount of mechanical work was dissipated in friction by a paddle wheel running in water. The rise in the temperature of the water indicated the amount of heat resulting from the dissipation of the work energy. As might be expected, the first experiments did not accurately establish the relation between heat and work units. In fact, it was many years later that Rowland, with highly refined apparatus, determined the value 778 (very nearly) that is universally used to-day.

It is interesting to note that this relation between heat and work, commonly symbolized by the letter J, need not be determined necessarily by direct experiment, but may be computed from other properties of gases, by the relation

$$c_p - c_v = \frac{R}{J} \text{ (Art. 85)}$$

As a matter of fact, the value of J had been computed before Joule made his experiment in 1843. But the specific volumes of gases from which R is calculated, and the two specific heats, were quantities all of which had to be determined by direct experiment, and as none of them had been well established at that time, the results of the calculation of J were not very accurate.

The First Law of Thermodynamics is really a special statement of the general law of the Conservation of Energy, as applied to the two particular forms of energy, viz., heat and mechanical work.

154. Insufficiency of the First Law of Thermodynamics.—The First Law states that heat and work are convertible one into the other, but it gives no hint as to whether or not the conversion can be realized completely in both directions or with equal facility. Proceeding by the guidance of the First Law alone, the engineer in the design and operation of heat engines might be led to two fallacious conclusions:

First, he might reason that heat is always convertible into work; that the efficiency of a heat engine is of little ultimate importance, since whatever heat "got by" one engine could be converted into work in another engine. Work ultimately goes back into heat, and this heat could be reconverted into work and no one would ever need be anxious about our store of useful energy running out.

Second, he might be led to believe that 100 per cent thermal efficiency could be attained in a heat engine. For example, if a pound of coal possesses a heating value of 14,000 B.T.U., it would seem entirely possible that a perfect heat engine might be conceived that would deliver $14{,}000 \times 778 = 10{,}892{,}000$ foot-pounds of work in using that pound of coal.

155. The Thought Underlying the Second Law of Thermodynamics.—The conclusions supposed in the preceding article we know to be erroneous from general experience, notwithstanding the fact that they do not violate the First Law. In every heat engine that has been considered, a refrigerator has been found to be an indispensable element of the system. Of the entire amount of heat, H, supplied to a working substance by the source of heat, the utilizer can not convert 100 per cent into work. A portion of the amount supplied must be dumped, as heat, into the refrigerator in order to make possible a completion of the cycle, and continuous operation of the engine, by a restoration of the working substance to its initial state.

It is evident, therefore, that in order to make the study of thermodynamics a science broad enough to account for all the facts, a limitation must be formulated that will preclude such fallacies as have been indicated. The formulation of this limitation really constitutes the Second Law of Thermodynamics.

156. The Second Law of Thermodynamics.—The Second Law of Thermodynamics may be stated thus: *Of a given quantity of heat, H, applied to a working substance, only a portion can possibly be converted into work; the remainder*

continues as heat. The maximum portion that can be converted into work is called the *Available Heat*, and is measured by the expression $\left(\frac{T_1-T_2}{T_1}\right)H$; the remainder is *Unavailable Heat*, and is measured by the expression $\left(\frac{T_2}{T_1}\right)H$; where T_1 and T_2 are the highest and lowest absolute temperatures furnished in the heat engine system.

157. Derivation of the Expressions $\frac{T_1-T_2}{T_1}$ and $\frac{T_2}{T_1}$.—The fact of the First Law can be established by direct experiment. The quantitative expressions, $\frac{T_1-T_2}{T_1}$ and $\frac{T_2}{T_1}$ of the Second Law, unlike the value of J of the First Law, must be derived analytically, although the foundation of the analysis rests upon two general statements of universal experience. (See Arts. 158 and 165.) Sadi Carnot, a Frenchman, who lived from 1795 to 1836, was the first to make this analysis, and determine the fractions that express the proportionality of available and unavailable energy of a given supply of heat.

Carnot reasoned that if a perfect heat engine could be conceived, the efficiency of such a heat engine would represent the proportion of available heat in the total quantity supplied. He had first therefore to attempt to conceive a cycle for a perfect heat engine, and he stated that the essential requirement of such a cycle is that it must be reversible. Inasmuch as a cycle consists of a series of operations, it follows that a reversible cycle is one that is composed of a series of reversible operations.

158. Reversible and Irreversible Operations.—In defining a reversible operation it is necessary to state a fundamental fact. *Heat cannot pass directly from a body of given temperature to one of a higher temperature.*

The proof of this statement rests upon universal experience. No one has ever been able to effect the transfer of heat from a cool body to a hot one, by placing them in direct

contact. The statement is accepted as true because it has never been disproved.

In discussing this question of reversibility, imagine a complete heat-engine system, consisting of a source, refrigerator, working substance, and heat utilizer, effectually isolated from the outside world so that there is no energy communication whatever. Certain operations can then be imagined to take place, and the effects upon the energy distribution of the system observed.

159. Direct Transfer of Heat From a Hot Body to a Cold One.—If the heat source be placed in direct contact with the refrigerator, there will occur a heat transfer from source to refrigerator. But the heat so transferred cannot be restored to the source by direct contact. It could be replaced only by means of some sort of a heat pump—a refrigerating machine. However, such a machine would call for an expenditure of mechanical energy, which in turn would necessitate a withdrawal of more heat from the source. By no possible means could the heat that has passed directly from the source to the refrigerator be restored to the source, leaving the system as it was in the beginning, without interchanges of energy with bodies outside of the system. Hence the passage of heat directly from source to refrigerator, or more generally, from a hot body to a cooler one, is an example of an *irreversible process.*

160. Isothermal Expansion.—If the container of the working substance at some intermediate temperature be placed in contact with the source, heat will flow into the working medium, and the operation is irreversible. If, the temperature of the working medium is only infinitesimally less than that of the source and is maintained so, there will yet be a transfer of heat from the source. In order to maintain the temperature of the working substance at the constant minute lower level than the source, the heat received must be passed on. If it is passed on by doing work, on a piston, for example, the operation is

isothermal. The work done upon the piston may be stored up in the mechanical form in the compression of a spring, the elevation of a weight, or the whirling of a fly-wheel. At the end of the forward operation, the stored mechanical energy can be called back and applied to the piston which can be made to compress the working medium back into its original restricted volume, pushing the heat of compression back into the source at the same constant temperature or infinitesimally higher. After the forward and reverse operations, it is found that the source possesses exactly the same amount of heat as it did originally; the working substance is unchanged; and no energy is in the mechanical form. The entire energy system is as it was in the beginning. The operation described is a *reversible process.*

161. Adiabatic Expansion—Throttling.—In an adiabatic change, the working substance is in communication with the heat utilizer only; it is wholly out of connection with either source or refrigerator. If expansion takes place with a gradual and orderly transfer of energy from the intrinsic heat stored in the gas to mechanical work of the piston, the operation is an adiabatic expansion at constant entropy, and is reversible. The work that is done upon the piston is sufficient to compress the gas in a reverse operation, and restore it to its original state.

The term adiabatic alludes to a process in which no heat, as heat, is supplied or rejected to the working substance. With this thought in mind, an adiabatic operation may be reversible or irreversible to any degree. The constant entropy change described above is a completely reversible operation. As an example of an adiabatic operation that is completely irreversible, refer to Joule's experiment (Art. 97). After opening the valve, and allowing the air to escape from the high-pressure vessel to the low, there is no means of reversing the process and reconcentrating the air back into the one vessel at its original pressure, without assistance from outside the

system. The operation is one of throttling, and throttling is an irreversible process.

It should be noted here again that the term " adiabatic," when used unmodified, almost always alludes to the reversible or constant entropy operation.

162. Mechanical Illustrations of Reversibility and Irreversibility.—Imagine an inclined plane connecting an upper floor of a building with the basement. There is a car that runs on the plane, and there is attached to one end of the car, by a suitable hook, a rope running over a pulley beyond the head of the incline. A weight is fastened to the other end of the rope, and suspended vertically from the pulley. The connection between car and pulley is such that this section of the rope is parallel to the incline. All the mechanisms are assumed to be frictionless. There is a supply of sand bags or other weights on the upper floor that can be loaded upon the car. The weight of the load on the car is available for doing work upon the weight that the rope raises vertically.

Now let sand bags be loaded on the car in just sufficient amount to lift the vertically moving weight. The slightest excess of pressure applied by the operator causes the system to function. When the car reaches the bottom of the incline, the potential energy of position originally possessed by the car and its load has been transferred to the weight, which was the objective sought.

The process can now be reversed, if desired. The operator can cause the elevated weight to pull the car with its load back to its original position. His energy system is then in exactly the same condition as it was at first. He need not call in any assistance from outside sources of energy. The descent of the loaded car is a reversible operation.

Suppose that somewhere along the inclined plane there is a step—an abrupt descent. Imagine the loaded car descending again, doing work upon the weight. When it arrives at the bottom of the incline, having successfully

dropped at the step, can the operator retrace the process, causing the elevated weight to bring the car and its load back to the top of the incline? He cannot do so without resorting to an outside source of energy for assistance. The original operation is irreversible. There has been a departure from the orderly state of balance between force and resistance and at that point a loss in availability of energy resulted.

Again, imagine an isolated energy system consisting of a tank of water elevated above the surface of a pond. For the purposes of illustration, it may be assumed that water can get from the elevated tank to the pond in three ways; through a short piece of pipe with a valve in it, attached to the tank; by leakage through cracks or holes in the tank; and through a water turbine of 100 per cent efficiency.

If the valve in the short piece of pipe is opened, water will spill from the tank to the pond. The process is obviously an irreversible one, because, without help from the outside, the spilled water can never be restored to the tank. The leakage of water from the tank is an irreversible process for the same reason. The latter operation differs from the former in that it is not controlled. It corresponds exactly to the thermal process of " radiation."

If the water is directed through the perfect turbine, the power developed may be stored up, and recalled later to operate the turbine as a pump. The power so developed is sufficient to reverse the original process, and restore the used water to the tank, leaving the energy system as it was originally.

163. An Irreversible Operation Means Loss of Available Energy.—Every energy system may be viewed with respect to its useful power possibilities. There will be found the working substance; the energy source, or high-head supply; the low-head repository of unavailable or wasted energy, known as the refrigerator in a thermal system; and the energy utilizer. The high-head energy of the

source is eager to break forth, and must be confined. If its escape from confinement is properly directed, useful results may be enjoyed from its willingness or availability. On the other hand, if the escape is improperly directed, or uncontrolled, not only is availability of energy destroyed without rendering any useful purpose, but the action may be accompanied by the most disastrous results, as for example, the failure of a dam or the burning of a building.

From the foregoing illustrations of reversibility and irreversibility, and many others that will come to mind, it is obvious that energy may proceed from a high-head source to a low-head repository in only two ways, viz., by a reversible or an irreversible operation. A reversible operation is one in which there is maintained a perfect balance between a yielding resistance and a motive force. There is an orderly and continuous transfer of energy. The potential energy of the loaded car on the inclined plane is gradually and smoothly transferred into potential energy of the vertically lifted weight. The water, passing from the elevated tank, through the perfect water turbine, to the pond, gradually imparts its potential energy to the wheel sinking without splash or commotion from one level to the other.

In an irreversible operation, there is usually a disordered, discontinuous, chaotic transfer of energy, accompanied by secondary transformations. When the loaded car drops at the step in the incline, the motive force exceeds the resistant force, and the excess becomes manifested in a kinetic energy of the masses. When the car bumps upon the bottom of the step kinetic energy is suddenly destroyed in impact, which results in the creation of heat. Some of the energy that should have gone into elevating the weight has found its way into unavailable heat. When water from the elevated tank is allowed to spill into the pond, either by the opening of the valve, or by leakage, there is similarly, first, an unresisted transformation of potential energy into the kinetic form, and then a further trans-

formation into heat. The whole operation is accompanied by disturbance, commotion, or splash. Again, the potential energy of the water, that might have been directed into some useful effect, now appears as useless heat. Parenthetically, it may be noted that the step of transforming potential energy into the kinetic form does not necessarily imply irreversibility. Kinetic energy is a mechanical form of energy, and as such is capable of being conserved and put to useful purpose, provided appropriate means—a suitable engine—is at hand for the purpose. In the illustrations used above, there were no suitable means of utilizing the kinetic energy, so it was inevitable that it should waste itself in impact—which is really a frictional effect.

The purely thermal irreversible process of direct passage of heat from a hot body to a cold one is analogous to the spilling of the water into the pond. Temperature is the heat head that corresponds to the gravity head of the water system. There is what may be called, having in mind the picture of the tank of water, a "thermal splash."

164. The Carnot Cycle.—From what has been said about reversibility and irreversibility, it is apparent that the reversible operation is the ideal one for a heat engine. An irreversible process is always accompanied by a transfer of available energy into unavailable heat. Hence it is clear that reversible operations, although ideal, and therefore impossible of actual attainment, must constitute the cycle of the ideal or perfect heat engine.

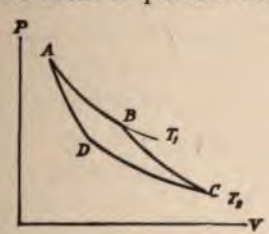

FIG. 100.

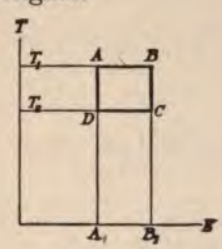

FIG. 101.

Carnot's cycle is represented in Figs. 100 and 101. Starting at state A, the working substance receives heat

along the isothermal AB, at the temperature of the source, T_1. At B, the working substance is cut off from the heat supply, and adiabatic expansion takes place to C, at which point the temperature has been lowered to T_2, the temperature of the refrigerator. From C the working substance is compressed isothermally to D, and thence adiabatically to the initial state A.

All of the operations are reversible. During AB and BC, heat is changed into work; during CD and DA, the working substance is restored to its initial state, partly by having work done upon it, and partly by cooling. Heat is supplied during the change AB, and is rejected during CD.

Referring to Fig. 101, the heat supplied is represented by the area A_1ABB_1, and the heat rejected, by B_1CDA_1. Hence:

$$\text{Efficiency of the cycle} = \frac{A_1ABB_1 - B_1CDA_1}{A_1ABB_1} = \frac{T_1 - T_2}{T_1}$$

Let $H = A_1ABB_1 = \text{heat supplied.}$

$$\text{Then the heat utilized} = \left(\frac{T_1 - T_2}{T_1}\right)H = ABCD$$

$$\text{and the heat rejected} = \left(\frac{T_2}{T_1}\right)H = B_1CDA_1$$

The efficiency of the Carnot cycle is that of the perfect or ideal heat engine. By means of it the heat quantity $H = A_1ABB_1$ (Fig. 101) is divided into the two fractions

$$\text{Available Heat} = ABCD = \left(\frac{T_1 - T_2}{T_1}\right)H$$

and

$$\text{Unavailable Heat} = B_1CDA_1 = \left(\frac{T_2}{T_1}\right)H$$

It remains to be proved that the efficiency of the Carnot cycle is the highest possible for any heat engine.

165. The Carnot Cycle Represents the Highest Possible Efficiency.—In order to prove that his cycle (or any

reversible cycle) represents the highest possible efficiency of a heat engine, it was necessary for Carnot to build upon another fact, which like that of Art. 158, is based upon general experience. This fact may be stated as follows:

In a segregated energy system, no change that can take place will increase the quantity of available energy. Imagine two heat engines, R and X (Fig. 102), of the same horse-power, operating between the same source, S, and refrigerator Z. Let R be a reversible engine, while X operates upon some cycle more efficient than that of R. X is arranged to drive R, so that R becomes a heat pump or refrigerating machine. The system is segregated from any outside communication, and possesses a certain amount of available energy, represented by the amount of heat in S, and its tendency to flow toward Z.

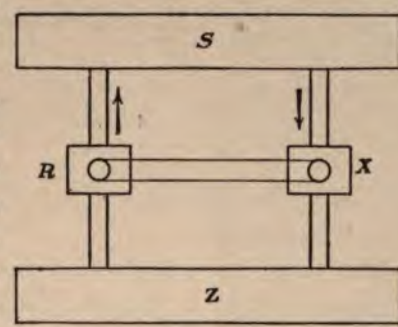

FIG. 102.

Now let the system be put into operation. X, being supposed to be more efficient than R, will take less heat from the source than would R if R were running direct and developing the same amount of power. Or, operating with R reversed, X will take less heat from the source than R will deposit in it. As a consequence, after a given time of operation, it would be found that the amount of heat in S was greater than it was in the beginning, which would mean that the available energy of this segregated system had been increased. As this is contrary to the facts of experience, it is to be concluded that the assumption of an engine of greater efficiency than a reversible engine R is an absurdity. Hence a reversible engine represents the highest possible efficiency for a heat engine, and the definite expression of Art. 156 to show the maximum pos-

sible portion of a given quantity of heat supplied, that might be converted into work, is established by the expression derived for the efficiency of a reversible engine in Art. 164.

$$\text{Heat supplied} \quad = H$$

$$\text{Amount available} = \left(\frac{T_1 - T_2}{T_1}\right) H$$

$$\text{Amount unavailable} = \left(\frac{T_2}{T_1}\right) H$$

where T_1 and T_2 are the absolute temperatures of source and refrigerator.

It is to be noted that these expressions are derived without regard to the properties of any substance. They are therefore perfectly general and independent of any working medium.

166. Other Reversible Cycles.—There are other possible reversible cycles besides that of Carnot. The ideal cycle of the Stirling Hot Air Engine (Arts. 148 and 149), making use of the idea of the heat regenerator, is an example. The working substance is passed bodily through a heat regenerator—an element consisting of a checker-work of some substance, with a considerable heat storage capacity. As the charge of gas at temperature T_1 enters the first section of the checker-work, it meets with a temperature only slightly less than its own, and imparts some of its heat to the checker-work. It then proceeds to the next zone at a slightly reduced temperature, and again meets with a temperature a little less than its own, where it imparts more of its heat. This process continues until the gas finally emerges from the regenerator at the temperature T_2, and in no case, theoretically, has there been more than an infinitesimal drop in temperature between the two bodies, gas and checker-work. The operation is ideally reversible.

Another example of a reversible cycle is that alluded to in Art. 50, where it was suggested in connection with Fig. 24, that the condensate (feed water) might be returned by way of a jacket around the barrel of the turbine in such a manner that the water at rising temperature would always be meeting surfaces in contact with steam at a trifle higher temperature, which would constitute essentially a reversible operation. The practical difficulty with this, as indeed with all reversible processes involving the transfer of heat, is the slowness of heat flow with small differences of temperature.

CHAPTER XV

THE DECREASE OF AVAILABLE ENERGY

167. Available Energy is Continually Decreasing.—The total amount of heat energy possessed by any system, or to extend the idea, possessed by that section of the Universe that yields itself to our study, remains constant. All transformations of energy from heat into other forms are but transient and almost momentary. However, although the total amount of heat energy is unchanged, the proportion of available heat—heat that may possibly appear as mechanical work—is continually decreasing, and as far as can be concluded from present observations, we are traveling toward a point of time on this planet when mechanical motion will cease to be. "In a segregated energy system, no change that can take place will increase the quantity of available energy." (Art. 165.) A reversible process can only leave the available energy unchanged; any other will result in a loss of available energy. A reversible operation is an ideal one and consequently never capable of attainment. Actually, therefore, every operation that takes place is an irreversible process, accompanied by a reduction in the amount of available energy and a corresponding increase in unavailable energy.

168. Illustration of the Continual Decrease of Available Energy.—As an example to show how every change that takes place helps to render some energy unavailable, a pictorial history of the heat of a pound of fuel is presented in the succession of figures that follow (Figs. 103 to 107, incl.) wherein the energy is shown in its various forms and substances from combustion of the fuel in the furnace of a

steam-power plant to final dissipation into the atmosphere in the form of completely unavailable heat.

For the example, most of the conditions of the power plant of Fig. 1 will be used. The heat of a pound of carbon as fuel will be considered rather than that of a pound of coal, to avoid the complication of giving attention to the ash.

The figures are all drawn to the same scale, so that temperatures and entropies at various stages are correctly represented by the coordinates of the figures, and heat quantities are proportional to the areas.

169. The Heat of Combustion—Zero Air Excess.—The heating value of a pound of carbon is 14,400 B.T.U. Assume both the carbon and the air with which it is to be burned to be at a temperature of 70° before combustion.

$$C + O_2 = CO_2$$
$$12 + 32 = 44$$

Each pound of carbon will require $32 \div 12 = 2.67$ pounds of O. The oxygen of air constitutes 23 per cent of the weight of the air.

$2.67 + .23 = 11.6$ pounds of air required to burn one pound of carbon.

The heat of combustion, assuming no "radiation" losses from the furnace, will appear as sensible heat of the gases of combustion, of which there will be 12.6 pounds for each pound of carbon. For the purposes of this illustration, the specific heat of the gases of combustion may be considered to be the same as that of air. In any case, there is but little error in this assumption because nitrogen constitutes the predominant element of both air and the products of combustion; and CO_2 is not very different in its properties from oxygen which it replaces.

Let t = temperature of combustion (°F.);
c_p = mean specific heat at a constant pressure;
W = weight of gas corresponding to one pound of fuel.

Then $$\overline{W}c_p(t-70) = 14{,}400$$

$$c_p(t-70) = \frac{14{,}400}{12.6} = 1142$$

From Langen's equation (Art. 132)

$$mc_p = 6.75 + .000667t$$

For air, the molecular weight, $m = 29$,

$$c_p = \frac{6.75}{29} + \frac{.000667t}{29}$$

$$= .233 + .000023t$$

Substituting

$$(.233 + .000023t)(t - 70) = 1142$$

$$.000023t^2 + .23139t - 16.31 = 1142$$

This is a quadratic, the solution of which results in the value, $t = 3670$.

Whence, $c_p = .318$

The increase of entropy of the 12.6 pounds of substance can now be computed.

$$\text{Increase of entropy} = 12.6 \times .318 \times \log_e \frac{3670+460}{70+460} = 8.23.$$

The entire heat quantity, 14,400 B.T.U., is represented by the area A_1ABB_1, Fig. 103. This heat quantity will raise the temperature of 12.6 pounds of air (or products of combustion) from 70° to 3670° F. The lowest available temperature is 70°—the temperature of the atmospheric air. It is therefore the refrigerator temperature. The area ABC above the 70° line represents the maximum possible portion of the 14,400 B.T.U. that can be changed into mechanical energy. The area A_1ACB_1 is the heat that is inherently unavailable. The area $ABC = 10{,}040$ B.T.U. Area $A_1ACB_1 = 4360$ B.T.U.

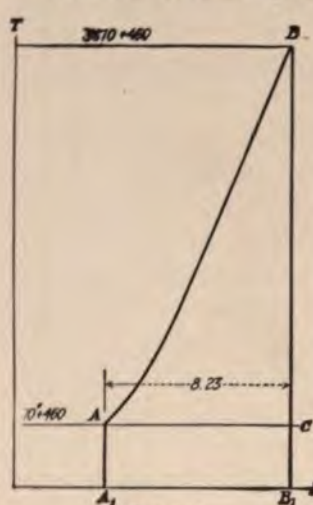

FIG. 103.

The available energy is 69.7 per cent of the entire heat of the furnace gases.

170. The Heat of Combustion—50 Per Cent Air Excess.—Practically, it is impossible to burn fuel, particularly solid fuel, with no air excess. Assume that the furnace of the plant, Fig. 1, is using 50 per cent excess of air. $(11.6 \times 1.50) + 1 = 18.4$ pounds furnace gas per pound of carbon.

Using Langen's expression for the relation between specific heat and temperature, as in the preceding article, it is found by method of Art. 169 that $c_p = .295$ and $t = 2720°$ F.

The increase of entropy in heating 18.4 pounds of substance from 70° to 2720° $= 18.4 \times .295 \log_e \frac{2720+460}{70+460} = 9.72.$

In Fig. 104 A_1ADD_1 represents the 14,400 B.T.U. developed by the combustion of the one pound of carbon. The area is of the same general shape as that of Fig. 103, but it is to be noted that it is wider, and not so tall.

The available heat,

$$ADF = 9250 \text{ B.T.U.}$$

The unavailable heat,

$$A_1AFD_1 = 5150 \text{ B.T.U.}$$

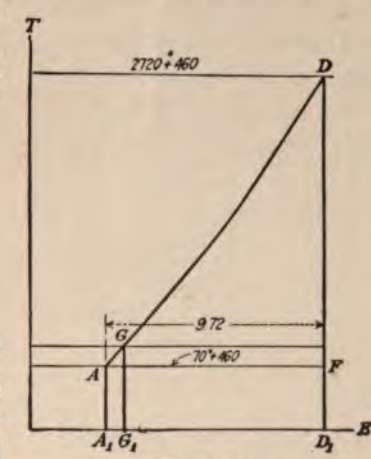

FIG. 104.

The available energy is now only 64.2 per cent of the entire heat of the furnace gases; 5.5 per cent more of the heat of the fuel has been rendered unavailable because of the necessity of burning with excess air.

171. Stack Loss.—In the power plant of Fig. 1, the furnace gases are represented as entering the stack at 300°, which is an unusually low temperature, resulting from the installation of an economizer. Taking the specific heat

of the stack gases to be .241, the heat that will be given up to the atmospheric air, in cooling the furnace gases back to 70°, is the amount of heat lost from the plant through the chimney per pound of fuel.

Heat lost to stack $= (300 - 70) \times .241 \times 18.4 = 1020$ B.T.U.

$14{,}400 - 1020 = 13{,}380$ B.T.U.—the heat that will be transferred to the steam per pound of carbon.

The area A_1AGG_1 (Fig. 104) represents the stack loss.

The area G_1GDD_1 represents the heat transferred to the steam.

172. The Heat in the Steam.—In the plant of Fig. 1 steam is delivered to the main unit at 190 pounds abs. and 150.4° superheat. The heat required per pound of steam, with hot-well temperature at 90°, is 1223.5 B.T.U.

$13{,}380 \div 1223.5 = 10.94$ pounds of steam produced for each pound of carbon burned.

The increase of entropy from water at 90° to steam at 190 pounds pressure and 150.4° superheat is 1.53 units per pound of steam, or a total of 16.74 units corresponding to the one pound of carbon. The heat received by the steam is pictured by the area H_1HJKLL_1, Fig. 105. The recep-

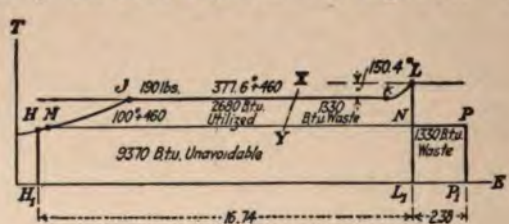

FIG. 105.

tion of heat by the steam is represented by the line $HJKL$. While the steam is receiving heat, a corresponding quantity of furnace gas is losing heat. Hence the appearance of the heat in the steam is to be associated with the disappearance of heat from the gases. The operation $HJKL$, Fig. 105, is simultaneous with the cooling operation DG, Fig. 104.

The lowest attainable temperature of the actual steam plant is 100°, in place of the true refrigerator temperature

of 70°. The available heat, measured above the 100° line, is 366.4 B.T.U. per pound of steam (Art. 45) or 4010 B.T.U. per pound of carbon. The unavailable heat is 13,380 − 4010 = 9370.

The ratio of available heat now present, to the entire heat of the one pound of carbon, is 4010 ÷ 14,400 = .278 or 27.8 per cent. The efficiency of transferring *heat* from furnace gases to steam is 13,380 ÷ 14,400 = .929 or 92.9 per cent —a very good performance. But viewed as an operation associated with a heat engine, where conservation of *available heat* is the real point, it is exceedingly wasteful. 36.4 per cent of the heat of the fuel is rendered unavailable by the irreversible operation of transferring it directly from a body of high temperature to one of low.

173. Transformation of Heat into Work.—With the acquisition of the heat by the steam, all is now ready for the climax in the series of events that are being pictured, viz., the transformation of heat into mechanical work in the heat utilizer. Observation of the actual performance of the power plant of Fig. 1 showed that out of each pound of steam, 245.2 B.T.U. were converted into work at the shaft of the turbine. (Art. 39.) The steam had 366.4 B.T.U. (Art. 172) available heat per pound when it entered the turbine. Hence, due to the inefficiency of the utilizer 121.2 B.T.U. per pound of steam (or 1330 B.T.U. per pound of fuel) have been allowed to slip from availability into unavailability. The exhaust steam gains in heat content by the amount of the turbine's waste.

The arbitrary line, XY, Fig. 105, is drawn to divide the area $MJKLN$ into the two sections to emphasize the idea that, in effect, the turbine takes the amount of the waste, 1130 B.T.U.; and disposes of it as unavailable heat in the exhaust steam, where it is represented by the area L_1NPP_1.

Out of 14,400 B.T.U. of heat in the fuel, 2680 B.T.U. or 18.6 per cent have finally been transformed by the heat engine into mechanical energy. And out of 9250 B.T.U.

possibly available (Art. 170), only 2680 B.T.U. are actually realized. 45.6 per cent of the entire amount of heat has been rendered unavailable by the wasteful processes described. Yet the performance of this plant is as good or better than the average steam plant.

174. The Heat in the Condenser Cooling Water.—Continuing with the history of the heat of the pound of carbon, consider next the transfer of the heat of the exhaust steam to the cooling water of the condenser. Water enters the condenser of Fig. 1 at 70° and leaves at 86°, a rise of 16°. The heat received by the water is the heat of the exhaust steam, 10,700 B.T.U., represented by the area P_1PMHH_1, Fig. 105.

$10{,}700 \div (1 \times 16) = 669$ pounds of cooling water required per pound of carbon.

The entropy of the water is increased by the amount

$$669 \times 1 \times \log_e \frac{86+460}{70+460} = 19.87.$$

The heat transferred to the cooling water is represented by the area Q_1QRR_1 of Fig. 106. The operation,

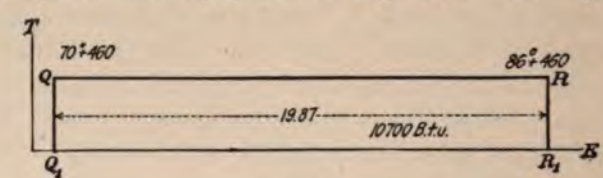

Fig. 106.

QR, of heating the water is simultaneous with the operation PMH (Fig. 105) of condensing and cooling the steam.

175. Dissipation of Heat to the Atmosphere.—The final operation to be pictured in this series is the disposal of all the heat of combustion of the pound of fuel into the surrounding atmosphere. The heat carried out of the plant by the condenser cooling water, and the heat of the furnace gases delivered to the stack are deposited here; and here also is to be found ultimately the heat equivalent of those 2680 B.T.U. (Fig. 105) that the turbine succeeded

in transforming into mechanical work. No matter to what diversified utilitarian purposes this energy is applied; whether it be to drive a tool through metal, or run a merry-go-round; to speed a train or propel a ship; to mine coal or weave a carpet; the opposition or resisting force by virtue of which the work is done, is in every case really the resistance of friction. Work is transformed back into heat that is picked up by the all-pervading atmosphere.

Of the 2680 heat units of energy transformed into mechanical work by the turbine of the plant of Fig. 1, only a part is finally applied to a useful purpose before sinking back into unavailable heat. Through whatever path of transmission or transformation the energy is conducted, it meets on every hand frictional resistances that exact their toll from an amount that is already discouragingly small when measured in terms of the original energy total.

The air of the atmosphere is of such vast heat capacity, that its temperature is hardly affected by such contributions as are being considered. Its reception of heat is practically an isothermal operation. The total heat of combustion of the pound of carbon is received by the air by way of three routes:

	B.T.U.
Heat from the stack gases	1,020
Heat from the cooling water	10,700
Heat resulting from the destruction of mechanical work energy	2,680
Total	14,400

The increase of entropy of the air for each of these several divisions is found by dividing the heat quantities by the absolute temperature, viz., $70+460=530$.

The final disposition of the heat is represented in Fig. 107.

S_1SUU_1 = heat lost in stack gases;

U_1UVV_1 = heat carried away by the cooling water of the condenser;

V_1VWW_1 = heat that has existed transiently as mechanical work;

S_1SWW_1 = entire heat of combustion of a pound of carbon, now dissipated in the atmosphere.

176. Entropy is Continually Increasing.—A comparison of the series of pictures, Figs. 103 to 107, discloses the gen-

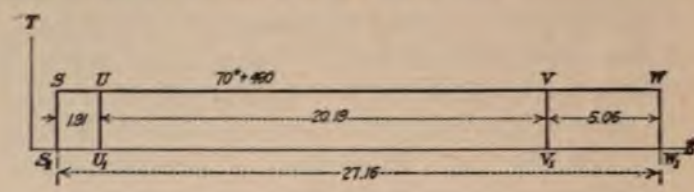

FIG. 107.

eral fact that after every operation the area representing the quantity of heat on the temperature-entropy plane is lower and wider. The temperature is diminished, and the entropy is increased.

Every change, except a reversible and therefore an ideal and actually unattainable one, reduces the available energy of a system. Likewise, every change, except a reversible one, results in an increase of the entropy of a system. Available energy is continually on the decrease; entropy is continually on the increase.

To show more clearly the fact that entropy increases with every operation except a reversible one, refer to Fig. 108. In nearly all cases a particular problem concerns itself with the changes that occur in the working substance only of a heat engine. However, if the net changes of a system are to be examined, it is apparent that the effects of a given change upon all the bodies of that system must be studied.

In Fig. 108 are pictured the effects of a change upon the source, S, the working substance, W, and the refrigerator, R, of a heat-engine system. Let the initial states of the source and working substance be the same, represented by point A. Now let W take heat from S, at constant tem-

perature. The heat received by W is of course just equal to that given up by S. The entropy of W increases to B and that of S decreases the same amount. AB is a reversible process, and the operation results in no change of the energy or the entropy of the system.

From B let the working substance expand to C, without supply or rejection of heat, but with frictional or throttling effect present as is the case for example when a substance expands in a nozzle that is not perfect. The state point of the substance is pushed to the right. The net change of

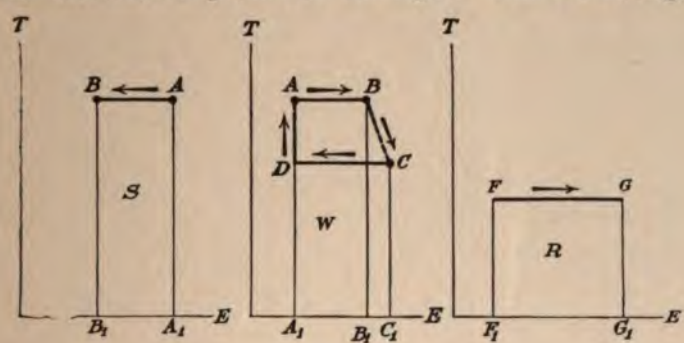

FIG. 108.

entropy of the system is B_1C_1 chargeable to the irreversible process BC.

Now let W be cooled at constant temperature to D, the refrigerator R withdrawing the necessary heat. The initial state of R is at F, a lower temperature than C. The heat quantity C_1CDA_1, withdrawn from W, is of course just equal to the heat quantity F_1FGG_1, the heat received by R. But since C_1CDA_1 is taller than F_1FGG_1, the latter must be wider. Hence FG, the growth of entropy of R is greater than CD, the diminution of entropy of W. The operation of cooling is an irreversible one, because there is a drop of temperature, and there results an increase of entropy.

Finally, let the cycle of W be closed by the adiabatic (constant entropy) compression from D to A. DA being a

reversible operation, no change of entropy of the system occurs.

Upon the completion of the cycle, the state of W is the same as it was initially. S has suffered a decrease of entropy, and R an increase. But the increase of entropy of R is greater than the decrease of S. Hence the net effect is an increase of the entropy of the system following the changes of an irreversible cycle.

CHAPTER XVI

THE FLOW OF FLUIDS

177. Working Media in Motion.—Heretofore, chief attention has been given to those cases in which the working substance of a heat engine is confined within a given space, and any transformation of heat into work has been accomplished by the direct application of the pressure of the medium to the overcoming of a yielding resistance. Some incidental attention has been given to the steam turbine, which is the best example of what may be called a kinetic engine. In the present chapter it is proposed to give consideration to the fundamental laws of the flow of fluids.

178. The Equation of the Continuity of Energy.—Let Fig. 109 represent the longitudinal section of a passage through which a gas is flowing from a higher pressure P_1 to a lower pressure P_2. The substance will enter the passage at section 1 with but little velocity and will emerge at section 5 with a higher velocity. In general there will occur a continual transfer of energy from the pressure, potential, or heat form to the kinetic or mechanical form. If suitable measuring instruments could be introduced at the several sections indicated, the transfer of energy would be manifested by a falling pressure, and temperature, and corresponding augmentation of velocity. If the flowing substance gets no heat from any outside source, and gives up no heat, during its passage through the channel, then it is evident that the total of its heat and kinetic energies must be the same at every section along the channel.

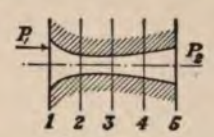

FIG. 109.

The heat energy is measured by the heat content, and the kinetic energy by the expression $\frac{V^2}{2g}$. We may then write:

$$JH_1+\frac{V_1^2}{2g}=JH_2+\frac{V_2^2}{2g}=\ldots JH_5+\frac{V_5^2}{2g}.$$

As written, the expression refers to one pound of the gas, H representing the heat content of the gas per pound, and $\frac{V^2}{2g}$, the kinetic energy per pound.

The above equation may be called the equation of the continuity of energy. It is a special case of the First Law of Thermodynamics, or the general law of the Conservation of Energy.

179. The Equation of the Continuity of Mass.—In Fig. 109, for any section designated, the following relation holds;

$$\overline{W}=\frac{AV}{S}$$

where $\overline{W}$ = weight passing the section per second;
A = area of the section in sq. ft.;
V = velocity of the gas at the section in ft. per sec.;
and S = specific volume of the gas at the section in cu. ft. per pound. (The change from V to S, to represent volume, is done to allow the representation of velocity by V.)

Although A, V, and S may have different values at other sections along the channel, $\overline{W}$ is the same at every point. Hence,

$$\overline{W}=\frac{A_1V_1}{S_1}=\frac{A_2V_2}{S_2}=\ldots\frac{A_5V_5}{S_5}$$

This equation may be called the equation of the Continuity of Mass.

180. Contour of a Nozzle Passage in Longitudinal Section.—A nozzle is an element whose primary function

is to convert pressure or heat energy into kinetic energy or velocity. The passage of Fig. 109 is a nozzle if its chief function is to generate velocity rather than merely to transfer a substance from one chamber to another.

The equation of the Continuity of Mass may be written thus:

$$\overline{W} = \left(\frac{V_1}{S_1}\right) A_1 = \left(\frac{V_2}{S_2}\right) A_2 = \ldots \left(\frac{V_5}{S_5}\right) A_5$$

As the gas progresses through the nozzle its pressure falls. The velocity increases and the specific volume increases. In order to allow the expansion to take place properly the area at any section must be such that it will accommodate the substance at whatever volume and velocity may prevail at that point. Or to put it the other way around, at any point along the nozzle a certain velocity and specific volume will exist, depending upon the area of the section. The quantity $\frac{V}{S}$ may be regarded as a coefficient of A. The manner in which both V and S vary depends upon the properties of the substance flowing. Hence the contour of the passage of a nozzle depends upon the nature of the substance flowing.

For example, consider a liquid—a substance whose specific volume undergoes no change with change of pressure. In traversing a nozzle, S will remain constant, while V increases. The value of the coefficient $\frac{V}{S}$ in the equation of the Continuity of Mass, is therefore continually increasing, and necessarily therefore A, the area of cross-section, should decrease. Case B of Fig. 110 illustrates the proper contour of longitudinal section of a nozzle suitable for a liquid.

Case C, Fig. 110, represents the correct contour for some hypothetical substance for which both velocity and specific volume increase at the same rate, so that their ratio is a constant at all points. The area of cross-section

should therefore be constant at all points, and the nozzle becomes a plain tube.

Case D is that of another hypothetical substance whose peculiarity is that, while both velocity and specific volume increase, the rate of specific volume increase is greater than that of the velocity, resulting in a divergent contour from the very entrance of the nozzle.

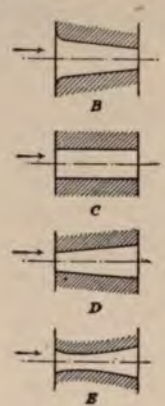

FIG. 110.

E is the general case of a nozzle suitable for an elastic fluid. It will be shown that for a gas, while velocity and specific volume both increase from the start, the velocity first increases faster than the specific volume, but after a certain critical point, specific volume increases more rapidly than velocity. Hence the value of $\frac{V}{S}$ first increases to a maximum, and then diminishes, necessitating a nozzle contour that is convergent-divergent in form.

181. Pressure in the Throat of a Nozzle—Example, Air Nozzle.—The general form of a nozzle for elastic media being convergent-divergent, there is a place of least cross-section between entrance and discharge ends. This place of least cross-section is called the *throat* of the nozzle. By reference to the equation of Art. 180, it is apparent that for a given set of initial conditions there must exist at the throat definite values for the velocity and specific volume, and consequently the pressure as well. If P_1 is the initial pressure, and P_t the throat pressure, it will be shown that, for air

$$\frac{P_t}{P_1} = .53$$

where

$$\frac{P_5}{P_1} < .53$$

This value will first be verified by a specific problem selected

at random, and later the general derivation will be given (Art. 183).

Example: Compressed air is furnished to a nozzle at 120 pounds per sq. in. abs. and 140° F. The nozzle expands the air adiabatically and discharges it at 20 pounds abs.: to show that

$$P_t = .53P_1 = .53 \times 120 = 63.6 \text{ pounds per sq. in. abs.}$$

Let calculations be made for a number of sections along the nozzle's length. The sections will be chosen at points to make the pressure intervals equal, say every 10 pounds. For any particular section, for example where the pressure is 50 pounds (the eighth point), the calculations will be as follows:

$$H_1 = c_p T_1 = .241 \times (140 + 460) = 144.6 \text{ B.T.U.}$$

Expanding adiabatically from 120 pounds and 140°, the temperature, $T_8 = 467°$ abs. (Art. 90).

$$H_8 = c_p T_8 = .241 \times 467 = 112.5$$

The initial velocity, where the pressure is 120 pounds, is practically zero. Hence

$$JH_1 = JH_8 + \frac{V_8^2}{2g}$$

$$\frac{V_8^2}{2g} \div 778 = H_1 - H_2 = 144.6 - 112.5 = 32.1 \text{ B.T.U.}$$

$$V_8 = 1268$$

At point 8, the pressure is 50 pounds and the temperature 467° abs. The specific volume can be computed

$$PS = RT$$

$$S = \frac{53.34 \times 467}{50 \times 144} = 3.46$$

$$\frac{V_8}{S_8} = 1268 \div 3.46 = 367$$

If it be assumed that one pound per sec. is passing through the nozzle

$$A_8 = 1 \div 367 = .00272 \text{ sq. ft. or } .392 \text{ sq. in.}$$

In like manner the computations may be made for each section, and results tabulated as below:

Point	Pressure	Temperature Abs.	Heat Content B.T.U.	Available Energy B.T.U.	Velocity, V.	Specific Vol. S.	$\frac{V}{S}$	A, sq. in.
1	120	600	144.6	0	0	1.85	0	
2	110	585	141.0	3.6	424	1.97	215	.67
3	100	569	137.1	7.5	613	2.11	291	.49
4	90	552	133.0	11.6	762	2.27	336	.43
5	80	534	128.7	15.9	892	2.47	361	.399
6	70	514	123.9	20.7	1018	2.72	374	.385
7	60	492	118.6	26.0	1141	3.03	377	.382
8	50	467	112.5	32.1	1268	3.46	367	.392
9	40	438	105.6	39.0	1397	4.05	345	.42
10	30	403	97.1	47.5	1542	4.97	310	.46
11	20	360	86.8	57.8	1701	6.66	255	.56

The values of velocity, specific volume, $\frac{V}{S}$, and area are plotted against the diminishing pressure, in Fig. 111.

It is seen that the value of $\frac{V}{S}$ increases to a maximum and then decreases. The pressure at which the maximum occurs is 63.6 pounds. At that point the area of the nozzle is a minimum. That is to say, at the throat of this nozzle, delivering air, the pressure is 63.6 pounds or $.53 \times P_1$. The example, selected at random, has verified the relationship stated at the beginning of this article.

The curves of Fig. 111 serve to show also how velocity at first increases faster than specific volume, but later the growth in volume becomes more rapid than that of the velocity, thus necessitating after the throat a growing area

to accommodate the "size" of the substance. It is to be noted that, although in a general way horizontal distances are proportional to length of nozzle, that relation is not necessarily direct. The nozzle might be imagined to be made of rubber, so that it could be stretched or compressed, from end to end; or one part of the length stretched and another compressed in any combination imaginable. But regardless of what changes are made in this manner, the pressure, velocity, and specific volume of any point depend

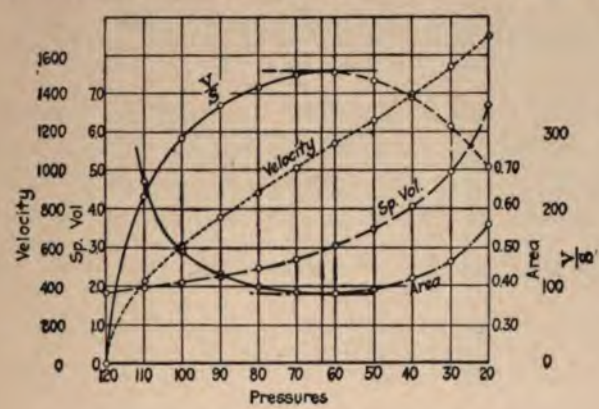

FIG. 111.

upon the area at that section, no matter whether the air travels an inch or a hundredth of an inch to drop 10 pounds.

182. Pressure in the Throat of a Steam Nozzle.— A specific problem, selected at random, might be worked out for a steam nozzle, following the same general method that was employed in the air-nozzle problem. The chief difference in the procedure would be in the use of the Steam Tables and Mollier chart to determine the properties at the various points, instead of computing them, as was done for air. The result would show that the pressure in the throat of the steam nozzle would be about .58 P_1.

The different value obtained for steam from that of air would lead to the conclusion that the relationship between throat pressure and initial pressure will be different for

each different gas, like other individual properties. The ratio $\frac{P_t}{P_1}$, for any gas, does in fact, depend upon the value of k.

$$\frac{P_t}{P_1}=\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$$

183. Derivation of the Relation $\frac{P_t}{P_1}=\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}$

The pressure in the throat occurs when $\frac{V}{S}$ is a maximum.

Let P, V, and S represent the pressure, velocity and specific volume at any point of progress along the nozzle's length. It is desired to obtain a suitable expression for V and S so that a differentiation can be made to ascertain the value of P when $\frac{V}{S}$ is a maximum.

$$\frac{V^2}{2g}=J(H_1-H)=Jc_p(T_1-T)$$

In terms of pressures and volumes

$$\frac{V_2}{2g}=\frac{k}{k-1}(P_1S_1-PS) \quad \text{(Art. 111)} \quad . \quad . \quad \text{(A)}$$

Where S = specific volume

$$PS^k=P_1S_1^k \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(B)}$$

$$\left(\frac{S}{S_1}\right)^k=\frac{P_1}{P}$$

$$S=S_1\left(\frac{P_1}{P}\right)^{\frac{1}{k}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(C)}$$

Substituting (C) in (A) and transposing $2g$,

$$V^2=\frac{2gk}{k-1}\left[P_1S_1-PS_1\left(\frac{P_1}{P}\right)^{\frac{1}{k}}\right] \quad . \quad . \quad . \quad \text{(D)}$$

Multiplying the second term in the brackets, of equation D by $\frac{P_1}{P_1}$

$$PS_1\left(\frac{P_1}{P}\right)^{\frac{1}{k}}\frac{P_1}{P_1}=P_1S_1\left(\frac{P_1}{P}\right)^{\frac{1-k}{k}}$$

Let r = the ratio $\frac{P}{P_1}$.

Then

$$P_1S_1\left(\frac{P_1}{P}\right)^{\frac{1-k}{k}}=P_1S_1\left(\frac{1}{r}\right)^{\frac{1-k}{k}}=P_1S_1(r)^{\frac{k-1}{k}}. \quad . \quad . \quad . \quad . \quad \text{(E)}$$

Substituting (E) in (D)

$$V^2=P_1S_1\frac{2gk}{k-1}\left(1-(r)^{\frac{k-1}{k}}\right) \quad . \quad . \quad . \quad . \quad \text{(F)}$$

Let $P_1S_1\frac{2gk}{k-1}=C_1$ a constant

$$V^2=C_1\left(1-(r)^{\frac{k-1}{k}}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(G)}$$

Again $S=S_1\left(\frac{P_1}{P}\right)^{\frac{1}{k}}$

$$S^2=S_1{}^2\left(\frac{P_1}{P}\right)^{\frac{2}{k}}=S_1{}^2\left(\frac{1}{r}\right)^{\frac{2}{k}} \quad . \quad . \quad . \quad . \quad \text{(H)}$$

Combining (G) and (H)

$$\frac{V^2}{S^2}=\frac{C_1\left(1-(r)^{\frac{k-1}{k}}\right)}{S_1{}^2\left(\frac{1}{r}\right)^{\frac{2}{k}}}$$

Let $\frac{C_1}{S_1{}^2}=C_2$, a constant, and let $z^2=\frac{V^2}{S^2}$

Then $$z^2=C_2\left[1-(r)^{\frac{k-1}{k}}\right](r)^{\frac{2}{k}}$$

$$=C_2\left[(r)^{\frac{2}{k}}-(r)^{\frac{k+1}{k}}\right] \quad . \quad . \quad . \quad . \quad . \quad \text{(L)}$$

Differentiating

$$2zdz = C_2\left[\frac{2}{k}(r)^{\frac{2-k}{k}} - \frac{(k+1)}{k}(r)^{\frac{1}{k}}\right]dr$$

To determine the value of r for which $z\left(=\frac{V}{S}\right)$, is a maximum,

Let $$\frac{dz}{dr} = 0$$

$$\frac{2}{k}(r)^{\frac{2-k}{k}} - \left(\frac{k+1}{k}\right)(r)^{\frac{1}{k}} = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(M)}$$

and $$r = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(N)}$$

The value of r, thus obtained, shows the ratio of the pressure at the throat to the pressure at the entrance of the nozzle, that is to say, the ratio where $\frac{V}{S}$ is a maximum. Hence

$$\frac{P_t}{P_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(P)}$$

For air, $k = 1.40$ and $\frac{P_t}{P_1} = .528$.

For steam, initially dry and saturated $k = 1.135$ and $\frac{P_t}{P_1} = .578$.

The value of k for steam depends upon the initial quality of the steam. Zeuner has shown that k is decreased by .001 for each per cent of moisture, from its value of 1.135 for steam at 100 per cent quality. Thus at 95 per cent quality, $k = 1.130$.

The ratio of throat pressure to initial pressure, as determined analytically, has been verified by numerous experiments in which the pressures of the fluid at various points along a nozzle were actually measured.

184. Significance of the Throat Pressure Relation.— In order to make clear the significance of the relation between throat pressure and initial pressure, the sketch of Fig. 112 is introduced. A is a high pressure chamber; B a low-pressure chamber; and N a nozzle delivering a fluid, for example, steam, from A, at a pressure P_1, to B, at a pressure P_2. A is supplied by a steam line from a boiler. D is a valve in the discharge pipe line from B. This pipe line may be thought of as connecting with a condenser, so that by the manipulation of D, any desired pressure can be maintained in B, from a high vacuum up to an equality with P_1.

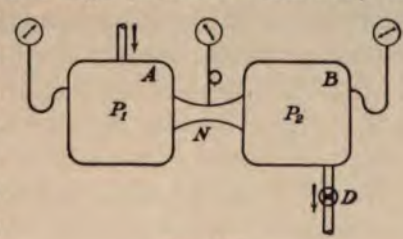

FIG. 112.

Imagine an experiment to be made in which the pressure P_1 is held at 150 pounds per sq. in. abs., while the back pressure is varied by means of D. Suppose D is wide open to begin with, when the back pressure P_2 on the nozzle is the lowest. D is then gradually closed until the passageway from B is completely shut off, when the pressure P_2 will become equal to P_1. How will the pressure P_t, in the throat of the nozzle be influenced by the variation of P_2?

The throat pressure has been shown to depend only upon P_1. For steam, $P_t = .58\ P_1$. In the case supposed, $P_1 = 150$, and therefore $P_t = 87$. It will be observed that the throat pressure gage remains constant at 87 pounds while the back pressure, P_2, is brought up from the minimum pressure to 87 pounds. But the moment that the back pressure exceeds 87 pounds, the throat pressure begins to rise, and from that point on, coincides with the back pressure.

As a result of this imaginary experiment, the following conclusions can be deduced; when the back pressure, on a steam nozzle, is less than 58 per cent of the initial pressure,

the throat pressure depends only upon the initial pressure (being 58 per cent of it) and is wholly independent of the back pressure: but when the back pressure is higher than 58 per cent of the initial pressure, the throat pressure then coincides with the back pressure; there is, in fact, no true throat pressure, because the back pressure is so high that it does not permit the steam to expand to that turning point in the relation of velocity and specific volume. The conclusions stated above with respect to the behavior of steam flowing through a nozzle (used as an example) apply equally well to all elastic fluids, with the corresponding value of $\frac{P_t}{P_1}$ introduced in place of .58.

185. Contour of Nozzle Passage as Affected by the Back Pressure.—A nozzle is an element that is to be situated between two conditions of pressure. In order that it may efficiently perform its function of generating velocity from pressure or heat energy, certain area relations must exist, conforming to the pressure conditions imposed. In general, these area relations result in a convergent-divergent contour of passage, similar to that of the example of the air nozzle of Art. 181. Referring to Fig. 111, in connection with that article, it will be observed that the area of the mouth of the nozzle (the discharge end) is .56 sq. in. The pressure in the nozzle at that point is 20 pounds, which is just the same as that of the chamber into which the nozzle is discharging. If the pressure in the discharge chamber were raised to 30 pounds by the manipulation of a valve in its exhaust pipe, as in Fig. 112, the nozzle should expand the air only to 30 pounds pressure, and the area at the mouth should be .46 sq. in. instead of .56 sq. in. as before. If the back pressure should be increased still further, the mouth area should be correspondingly decreased. The nozzle might be imagined to be cut off at successive points to make the mouth area agree with the back pressures imposed.

When the back pressure has been increased to 63.6

pounds, the nozzle would be cut off at the throat. The divergent section of the nozzle has entirely disappeared.

Up to this point, any change in back pressure does not project its influence backward into the nozzle, so to speak. None of the nozzle pressures are affected. Even if the mouth area of the nozzle were not changed to suit the back pressure, the nozzle would theoretically go on expanding the air just as it did before, leaving the fluid to adjust itself to the changed pressure condition outside, after it was discharged. For illustration, suppose the back pressure of the example represented by Fig. 111 were increased from 20 to 30 pounds, but the mouth area is left at .56 sq. in. Theoretically, the nozzle will go on expanding the air to 20 pounds, the kinetic energy of the air throwing it out into the region of higher pressure. Practically, this unbalanced relation between back pressure and mouth area, is found to be productive of influences that extend back into the nozzle and affect its efficiency detrimentally.

When the nozzle has been cut off at the throat, that is, when the divergent section is not present, any further increase in the back pressure above the true throat pressure will result in a readjustment of the pressures at the different sections of the convergent nozzle, to give a gradual drop from the initial to back pressure. It is no longer necessary to change the mouth area to agree with the back pressure in order to maintain efficient expansion of the fluid.

The conclusions arrived at in this article may be summarized as follows:

When the back pressure on a nozzle is less than the throat pressure, the nozzle must have the convergent-divergent shape, and the mouth area should correspond to the back pressure.

When the back pressure exceeds the true throat pressure, the nozzle is convergent only, and no necessary relationship exists between the mouth area and the back pressure, as far as the efficient production of velocity is concerned.

186. The Straight Nozzle.—The convergent nozzle for the condition of back pressure exceeding the true throat pressure is usually provided with a straight or non-divergent section of uniform area, as illustrated in Fig. 113.

The addition of the straight section in no way influences the performance of the nozzle except to direct the jet to better advantage. The velocity is all generated in the convergent section.

It is interesting to inquire how this nozzle would handle a fluid if the back pressure in the discharge chamber were less than the throat pressure. In the case of the example of Art. 181, how would the air behave in passing through this straight nozzle, initial and back pressures being 120 and 20 pounds respectively, as there specified? The expansion of the air to the throat—the section of least area—would be exactly the same as for the convergent-divergent nozzle. From this point on, however, it is impossible for the air to drop further in pressure in the nozzle. Referring to Fig. 113:

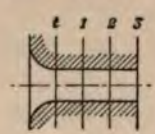

FIG. 113.

$$\frac{A_t V_t}{S_t} = \frac{A_1 V_1}{S_1} = \frac{A_2 V_2}{S_2} = \frac{A_3 V_3}{S_3}$$

or

$$\left(\frac{V_t}{S_t}\right) A_t = \left(\frac{V_1}{S_1}\right) A_1 = \left(\frac{V_2}{S_2}\right) A_2 = \left(\frac{V_3}{S_3}\right) A_3$$

But

$$A_t = A_1 = A_2 = A_3$$

Hence

$$\frac{V_t}{S_t} = \frac{V_1}{S_1} = \frac{V_2}{S_2} = \frac{V_3}{S_3}$$

There can be no change in the relationship between V and S. But as shown by the $\frac{V}{S}$ curve of Fig. 111, there

is only one pressure that can satisfy the relation of V and S that exists at the throat. Consequently, the throat pressure prevails throughout the length of the straight section. Upon its discharge from the nozzle, the pressure of the air drops instantly to the back pressure. The straight section of the nozzle allows no further augmentation of the velocity. Available energy that would have appeared as velocity in the discharge of a properly proportioned divergent nozzle, is dissipated in unresisted expansion outside of the straight nozzle.

187. Usual Shape of Nozzles.—As usually made, nozzles are considerably modified in contour from the general shape of Fig. 109. The convergent section is reduced in length until it is no more than a well-rounded entrance. The walls of the divergent portion are of straight-line elements, to facilitate manufacture. The efficiency of a simple nozzle of this kind, illustrated in Fig. 114, is found to be as good as that of nozzles of more elaborate form. In cross-section nozzles are circular, square, or rectangular. The length of the nozzle from throat to mouth is determined with some regard to the angle of divergence of the walls. The angle a should not be greater than 12°.

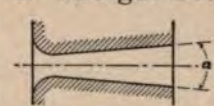

FIG. 114.

The nozzle of Fig. 114 would not allow a uniform pressure drop throughout its length. The drop would be very abrupt from entrance to throat, after which it would be much more gradual.

188. Weight of Discharge through a Nozzle.—Fliegner's Formula.—For any given nozzle operating between specified pressures, a set of curves could be constructed, like those of Fig. 111 with the addition that it would be necessary to plot an *area-ratio* curve to show the relation of areas at different points in terms of the area of some particular section, preferably the throat area, taken as unity. At any point chosen along the actual given nozzle, the area of section could be determined either by direct measure-

ment, or from drawings, and expressed as a ratio to the throat area. For any particular area relation, there are definite values for the specific volume and velocity, which could be read directly from the curves already constructed. Then from the relation

$$\overline{W} = \frac{AV}{S}$$

$\overline{W}$ could be computed.

The above method of procedure would involve a great deal of labor. When the back pressure is below the throat pressure, the throat pressure is known at once, and the velocity and specific volume at the throat can be computed and $\overline{W}$ can be determined by the equation

$$\overline{W} = \frac{A_t V_t}{S_t}$$

The computation of $\overline{W}$ can be still further simplified for the case where the back pressure is less than the throat pressure. Referring to Art. 183, the following equations were met with in the course of the derivation.

$$V^2 = P_1 S_1 \left(\frac{2gk}{k-1}\right)(1-r)^{\frac{k-1}{k}} \quad . \ . \ . \ . \quad \text{(F)}$$

$$S^2 = S_1{}^2 \left(\frac{1}{r}\right)^{\frac{2}{k}} \quad . \ . \ . \ . \ . \ . \ . \ . \quad \text{(H)}$$

where V is the velocity at any point along the nozzle;
S is the specific volume at that point;
P_1 is the initial pressure;
S_1 is the initial specific volume;
$r = \frac{P}{P_1}$. P is the pressure at the point where V and S occur.

If the throat section be considered, r then has a definite value for a particular gas. To simplify the work let the calculations be made for air, for which $r = .53$ and $k = 1.40$.

Then

$$r^{\frac{k-1}{k}} = (.53)^{\frac{k-1}{k}} = .833$$

$$r^{\frac{2}{k}} = \left(\frac{1}{.53}\right)^{\frac{2}{k}} = 2.48$$

$$\frac{2gk}{k-1} = \frac{2 \times 32.2 \times 1.40}{.40} = 225.4$$

Dividing equation (F) by equation (H), and substituting values computed

$$\frac{V_t^2}{S_t^2} = \frac{37.64\ P_1 S_1}{2.48\ S_1^2}$$

But

$$S_1 = \frac{RT_1}{P_1}$$

whence

$$\frac{V_t^2}{S_t^2} = 15.18 \frac{P_1^2}{RT_1} = .284 \frac{P_1^2}{T_1}$$

$$\frac{V_t}{S_t} = .53 \frac{P_1}{\sqrt{T_1}}$$

And

$$\overline{W} = A_t \frac{V_t}{S_t} = .53 \frac{A_t P_1}{\sqrt{T_1}}$$

As derived, A_t is the area of the throat in square feet and P_1 is the initial pressure in pounds per sq. ft. It is usually more convenient to express the area in square inches, and the pressure in pounds per sq. in. The substitution of these latter units for the former does not alter the constant, wherefore

$$\overline{W} = .53 \frac{a_t p_1}{\sqrt{T_1}}$$

where $\overline{W}$ = weight of air delivered per sec.;
a_t = area of the throat of the nozzle in sq in.;
p_1 = initial pressure pounds per sq. in. abs.;
T_1 = absolute initial temperature ° F.

This equation is known as Fliegner's formula for the discharge of air. Fliegner verified its accuracy by experiment. It is applicable when the back pressure is less than .53 of the initial pressure. It applies to nozzles and not to orifices.

189. Discharge of Steam through Nozzles.—Grashof's and Napier's Equations.—The same procedure may be followed in working out a rational formula for the flow of steam, except that in place of using the characteristic equation to furnish a substitution for S_1 in equation (H), Art. 188, it is necessary to use an empirical relation between pressure and volume of saturated steam. This relation as given by Zeuner, is

$$P_1 S_1^{1.0646} = PS^{1.0646} = C$$

The value of the constant C is about 487.5, when P is expressed in pounds per sq. in. and S in cu. ft. It can be verified by substituting specific values for P and S from the Steam Tables and solving for C.

The resulting equation for the flow of steam is

$$\overline{W} = .01654 a_t p_1^{.9696}$$

where $\overline{W}$ = the weight of steam discharged per sec.;
a_t = the area of the nozzle at the throat, sq. in.;
P_1 = the initial pressure in pounds per sq. in. abs.

This equation was first derived by Grashof. A somewhat more convenient modification is that suggested by Moyer.

$$\overline{W} = \frac{a_t p_1^{.97}}{60}$$

The most widely used formula for calculating the discharge of steam through a nozzle is an empirical expression by Napier, commonly called Napier's rule. Its simple

form makes it extremely convenient to apply under any circumstances.

$$\overline{W} = \frac{a_t p_1}{70}$$

Careful experimental investigation shows that the weight of steam delivered by a nozzle discharging against a back pressure, is not quite a straight-line function of the initial pressure, thus confirming the rational equation. The curve for Napier's rule, a straight-line function, crosses the experimental curve at two points; the maximum error by Napier's rule, within the ordinary range of pressure, is slightly over 2 per cent. The maximum error by Grashof's equation is about 1 per cent.

190. Discharge Formulas Summarized.—The most common and useful formulas for calculating the weight of discharge through nozzles are summarized below.

For air.

When the back pressure is less than 53 per cent of the initial pressure:

$$\overline{W} = .53 \frac{a_t p_1}{\sqrt{T_1}} \text{ (Fliegner)} \quad . \ . \ . \ . \ . \ . \ \text{(A)}$$

When the back pressure is greater than 53 per cent of the initial pressure, the following equation may be used:

$$\overline{W} = 1.06 a_t \sqrt{\frac{p_2(p_1 - p_2)}{T_1}} \text{ (Fliegner)} \quad . \ . \ \text{(B)}$$

where $\overline{W}$ = weight of air delivered in pounds per sec.;
a_t = area at throat or smallest section of nozzle, sq. in.;
p_1 = initial pressure, pounds per sq. in. abs.;
p_2 = back pressure, pounds per sq. in. abs.;
T_1 = absolute initial temperature, ° F.

For steam.

When the back pressure is less than 58 per cent of the initial pressure:

$$\overline{W} = .01654\, a_t p_1.^{9696} \text{ (Grashof)} \quad . \quad . \quad . \quad \text{(C)}$$

$$\overline{W} = \frac{a_t p_1^{.97}}{60} \text{ (Moyer)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(D)}$$

$$\overline{W} = \frac{a_t p_1}{70} \text{ (Napier)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(E)}$$

When the back pressure is greater than 50 per cent of the initial pressure.

$$\overline{W} = \frac{a_t p_2}{42}\sqrt{\frac{3(p_1 - p_2)}{2p_2}} \text{ (Napier)} \quad . \quad . \quad . \quad \text{(F)}$$

All the equations given are applicable to nozzles only, not to orifices.

The steam equations, as given, apply only when the steam is initially at 100 per cent quality. For other conditions, either superheated or wet, correction factors must be applied.

For superheated steam the following empirical factor is quite generally used:

$$\frac{1}{1 + .00065D}, \text{ where } D = \text{number of degrees superheat.}$$

For steam initially wet, the factor given below was deduced from experiments made in the Mechanical Engineering Laboratory of the University of Michigan.

$$\frac{1}{1 - .012m} \text{ where } m = \text{number of per cent moisture.}$$

191. Influence of Back Pressure upon Rate of Discharge.—Referring to Fig. 112, let another imaginary experiment be performed with the apparatus there described. Let a constant initial pressure of $P_1 = 150$ pounds

abs. be maintained in A. Assume that the experiment is made with steam, and that the exhaust pipe from B communicates with a surface condenser, so that not only may the back pressure in B be reduced to less than atmospheric, but the discharge steam can be condensed and its weight determined. The back pressure, P_2, is to be varied by the valve D, and the influence of the varied pressure upon the rate of flow is to be observed. Lay off the coordinates for a curve OAM, Fig. 115, to show the relationship between P_2 and $\overline{W}$.

First, let the valve D be entirely closed. Of course there will be no flow. With D slightly opened, the back pressure will fall a little, and some steam will be discharged. As the back pressure decreases still further the weight of discharge increases as indicated by the curve OA, until the back pressure has been reduced to .58 P_1. From that point on any further decrease in P_2 results in no augmentation of the weight discharged. No more steam will get through the nozzle with a vacuum of 28 in. in the chamber B than when there is a pressure of 60 pounds. When the back pressure falls below the throat pressure, it ceases to have any influence upon the weight of discharge.

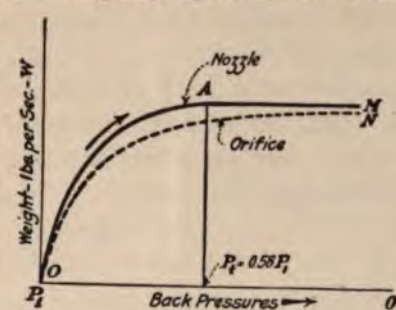

FIG. 115.

This peculiar and important fact is a part of the general phenomena associated with the rigid relationship that exists between throat and initial pressures. When the back pressure is greater than the true throat pressure, its influence is extended back into the nozzle to the place of least cross-section. This is true whether the nozzle has a divergent section beyond the throat, a straight section, or no length beyond the throat at all. The back pressure

replaces what may be called the natural throat pressure. Intermediate pressures between P_1 and P_2 automatically adjust themselves to the succession of cross-sections between entrance and end of the convergent portion of the nozzle, so that the drop is gradual, no matter what the value of P_2, so long as it is above the natural throat pressure. The back pressure therefore exerts a direct influence upon the flow, the variation in back pressure resulting in a variation in flow as already indicated by the section of the curve, OA. Formulas for computing the weight of discharge under the condition of back pressure higher than the true or natural throat pressure, must necessarily contain the value of the back pressure as a factor, as in equations (B) and (F), Art. 190.

When the back pressure is less than the throat pressure, it then has no influence upon the throat pressure, and of course does not affect the adjustment of pressures between entrance and throat.

$$\overline{W} = \frac{AV}{S} = \frac{A_t V_t}{S_t}$$

V_t and S_t being functions of P_t, which in turn is wholly independent of P_2, it follows that the weight of flow, $\overline{W}$, is also wholly independent of the back pressure.

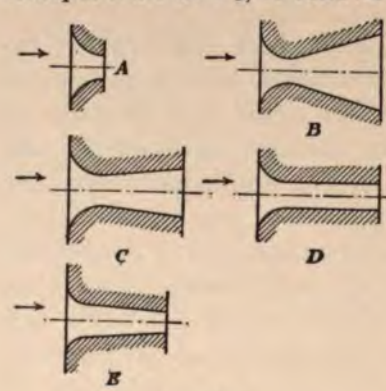

FIG. 116.

It may be stated that the weight of flow is independent of the shape or character of the section that lies beyond the throat of a nozzle,

In Fig. 116 let A represent a simple convergent nozzle, while B, C, D, and E represent four nozzles all having their entrance ends identical with A, while the sections beyond are different in each case. B shows a widely divergent

addition beyond the throat; *C*, a moderately divergent addition; *D*, a straight section; and *E*, a convergent section continuous with the entrance part. For the same conditions of pressure, the weight of flow through *A*, *B*, *C*, and *D* is the same. The different sections attached beyond the throat do not affect the weight of flow. In the case of *E*, the convergent section attached to the part *A* does not constitute an addition beyond the throat. The instant the additional convergent piece is attached, the small end of the new piece becomes the throat. The area of the throat has been altered, and, therefore, as a matter of course, the rate of flow is changed.

It must not be forgotten that a nozzle's primary function is to generate velocity from pressure or heat energy, and in the performance of this function, the proportioning of the divergent section or portion beyond the throat is all-important to efficient energy transformation.

192. Nozzle Calculations.—The throat area of a nozzle must be calculated with reference to the weight of substance to be discharged per unit of time; and the mouth area must be properly related to the throat area in order that the fluid may be completely expanded to the back pressure.

As an illustration, let the calculations be made for a nozzle to deliver 2400 pounds of steam per hour, receiving the steam at 125 pounds abs. pressure and 50° superheat, and delivering against a back pressure of 10 pounds abs. pressure.

Using Napier's rule, to determine the throat area a_t from the specified conditions:

$$2400 \div 3600 = \frac{a_t \times 125}{70} \times \frac{1}{1 + .00065 \times 50}$$

$$a_t = .385 \text{ sq. in.}$$

The mouth and throat areas are connected by the continuity equation

$$W = \frac{A_t V_t}{S_t} = \frac{A_m V_m}{S_m}$$

where A_t, and A_m = areas in sq. ft.

V_t and V_m = velocities in ft. per sec. and S_t and S_m = specific volume in cu. ft. at the throat and mouth sections, respectively. The throat pressure, $P_t = .58P_1 = .58 \times 125 = 72.5$

From the Mollier Chart,

$$H_1 - H_t = 1219 - 1172 = 47 \text{ B.T.U.} \quad q_t = 99.1$$

$$\frac{V_t^2}{2g} = J \times 47$$

$$V_t = \sqrt{2 \times 32.2 \times 778}\sqrt{47} = 223.7\sqrt{47} = 1535$$

$$S_t = 6.00 \times .991 = 5.946 \text{ cu. ft.}$$

The available energy of the steam from the initial conditions to the back pressure, as found from the Mollier Chart is:

$$H_1 - H_m = 1219 - 1032 = 247 \text{ B.T.U.}$$

The efficiency of the nozzle may be assumed to be 96 per cent, which means that 96 per cent of the available energy will be converted into kinetic energy.

$$V_m = 223.7\sqrt{.96 \times 247} = 3445.$$

With no loss, the heat content of the jet of steam discharged from the nozzle would be 1032 B.T.U. As a matter of fact, however, 4 per cent of the available energy, or 9.9 B.T.U. waste energy is added to the 1032 B.T.U. originally unavailable heat, making a total of 1041.9 B.T.U. The discharged steam is at a pressure of 10 pounds and its heat content is 1041.9 B.T.U. Its quality is found to be 89.8 per cent. The specific volume of dry steam at 10 pounds pressure, is 38.38 cu. ft.

$$S_m = 38.38 \times .898 = 34.47 \text{ cu. ft.}$$

Substituting the several results in the continuity equation

$$\frac{(.385 \div 144) \times 1535}{5.946} = \frac{(a_m \div .144) \times 3445}{34.47}$$

$$a_m = .994 \text{ sq. in.}$$

If the nozzle is to be circular in cross-section, the diameter at throat and mouth will be .700 in. and 1.126 in. respectively. The divergent portion of the nozzle will be conical, and the length will be determined by the requirement that the total angle of divergence of the walls shall not be greater than 12°. For the convergent portion of the nozzle, merely a well-rounded entrance to the divergent section is sufficient.

193. Flow through Orifices.—An orifice is a passage for flowing fluids in which the area is large with respect to the length, in contradistinction to a nozzle, wherein the length is large in comparison to the lateral dimensions of the passage. In general, the equations of flow for a nozzle will not apply to an orifice.

In Fig. 117 *A* and *B* represent orifices in thin plates. *A* has sharp edges, while *B* has a rounded entrance. When flow takes place the stream lines of the fluid really constitute a convergent nozzle, the true area of the throat of which is the *vena contracta* of the jet instead of the area of the orifice. If it were possible to measure the area of the vena contracta in a given case, it is probable that the substitution of that value for the term a_t in the flow formulas for nozzles would give fairly accurate results.

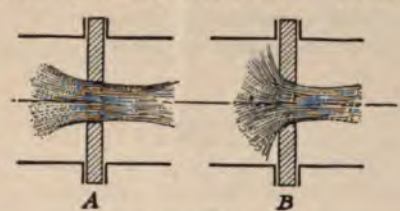

FIG. 117.

For an orifice there appears to be no strictly definite critical value of the back pressure, below which the latter ceases to manifest an influence upon the weight of flow. The situation is represented by the curve *ON* of Fig. 115, in which it is seen that reduction of back pressure, even down into the lower ranges, results in a slight increase in flow.

Problems

1. Steam is supplied to a nozzle at 120 pounds abs. and 100 per cent quality; the back pressure is 20 pounds abs. Considering adiabatic expansion, let calculations be made for pressures at 10-pound intervals, and the results set down in a table like that of Art. 181, except that in the third column " quality " will replace temperature. Plot the curves of Fig. 111, and ascertain the pressure at the throat of the nozzle and its relation to the initial pressure.

2. If ammonia is passed through a nozzle, what is the relation of throat pressure to initial pressure? Sulphur dioxide? Helium? (See Table of Art. 130.)

3. Compressed air at 97.5 pounds per sq. in. abs. pressure and 90° temperature, is discharged through a nozzle into the atmosphere. The diameter of the nozzle at the throat is .443 in. What weight of air in pounds per minute will be discharged? What is the delivery in cu. ft. of free air per minute?

4. What diameter of throat must a nozzle have in order to deliver a ton of steam per hour from a pressure of 140 pounds per sq. in. abs. and 98 per cent quality against a back pressure of 50 pounds per sq. in. abs.?

5. If initial steam pressure is 160 pounds abs. and quality 100 per cent; and the back pressure is 15 pounds abs., calculate the discharge in pounds per hour for a nozzle whose throat area is .35 sq. in. by Grashof's, Moyer's and Napier's equations respectively. Also calculate the discharge by the equation

$$\overline{W} = 3600 \frac{A_t V_t}{S_t}$$

and compare the four results.

6. What weight of air per minute will be delivered by a nozzle of $\frac{1}{2}$ in. diameter at the throat when discharging into the atmosphere from a pressure 10 pounds above atmospheric? The atmospheric pressure is 29.44 in. Hg. The initial temperature of the air is 75°.

CHAPTER XVII

KINETIC ENGINES. THE STEAM TURBINE AND THE INJECTOR

194. Kinetic vs. Direct Pressure Engines.—The steam turbine is the most familiar example of the kinetic engine, and the distinction between the kinetic and direct-pressure principles will be brought out by a comparison between the steam turbine and the steam reciprocating engine.

Work is the product of two elements, Force and Space.

$$\text{Work} = \text{Force} \times \text{Space}.$$

In the reciprocating steam engine, force is the result of the direct application of the pressure resulting from the thermal condition of the steam, to the yielding resistance of the piston.

In the steam turbine the force is the result of the pressure that is set up when a mass is hurled against a surface that can move in response to the impulse. The mass is the working fluid—the steam itself; the yielding resistance is the blade.

The distinction between the two principles is clear cut. The piston of the steam reciprocating engine receives no push from any stream of steam playing upon it; neither is the blade of a turbine propelled in any degree directly by the elastic pressure of the working substance.

The turbine heat engine necessitates two distinct operations in the transformation of heat energy into mechanical work:

a. The conversion of as much as possible of the heat energy of the working substance into kinetic energy of the working substance.

b. The communication of the kinetic energy of the working substance to the mechanism of the turbine whereby it may appear in useful form.

The first operation is performed by the nozzle of the turbine. It is strictly a thermodynamic problem, and has been discussed, in a general way, in the preceding chapter. The second operation is, in its theoretical aspect, a purely mechanical problem, involving the transformation of mechanical energy of one kind into another. However, on account of unavoidable losses, the transfer of the kinetic energy of the fluid into useful form is not complete, the residue reverting to heat again; hence the second operation is, practically, a thermodynamic one.

195. Types of Turbines.—There are two kinds of turbines, viz., the impulse and the reaction, depending upon the manner in which the energy is transferred from the energy of the jet to the rotor of the turbine.

A jet may be made to exert a push by directing it against a surface in its path, as, for example, the playing of a water hose upon a paddle wheel. The jet acquires its full velocity before any attempt is made to realize a force or push on the rotor of the machine. The direction of motion of the yielding resistance—the blade—generally speaking, is the original direction of the jet. What has been described is the *principle of the impulse turbine.*

Velocity of a jet can be imparted only by the application of an aggregate force sufficient to accelerate its component particles from a speed of little or nothing to the full velocity. This is what occurs in a nozzle. The force of acceleration is in the direction of the jet; its reaction is in the opposite direction. There is a backward push on the nozzle, whether the nozzle is permitted to move or not.

If the nozzle can move, work is done, and the *principle of the reaction turbine* is employed. The useful force or push on the rotor is exerted simultaneously with the creation of the velocity. The nozzle performs the functions of both the velocity generating element and the blades, of the impulse turbine.

196. Classes of Impulse Turbines.—The impulse turbine is capable of exemplification in a variety of combinations. It may be single- or multiple-pressure stage, and each pressure stage may have a single velocity stage, or several velocity stages.

In a single-pressure stage turbine, all the available heat energy possible is transformed into kinetic energy of the jet in one step. The kinetic energy thus created is then communicated to the rotor by means of one or more velocity stages.

In a multiple-pressure stage turbine, only a portion of the available heat energy is converted into kinetic energy by the first set of nozzles. The resulting velocity is absorbed in one or more velocity stages. A portion of the remaining available energy is then converted into kinetic energy by another set of nozzles, and the velocity is absorbed in corresponding velocity stages; and so on until the entire available energy is utilized. A multiple-pressure stage turbine is in effect a group of single-pressure stage turbines operating serially upon the working substance.

It is assumed that the reader is somewhat familiar with the mechanical arrangement of these different classes of turbine so that no further description or illustration of them will be given here. It is proposed to show in a general way the heat changes that occur in the steam, during its passage through the turbine.

197. Heat Changes in a Single-pressure Stage Turbine. For illustration, assume a single stage turbine to operate between the initial condition of 150 pounds abs. pressure and 98 per cent quality, and an exhaust pressure of one

pound abs., which corresponds to a vacuum of about 28 in. Assume also that the turbine is to have three velocity stages.

Fig. 118 represents the heat-entropy or Mollier heat chart. The initial state of the steam is found at A, the intersection of the 150-pound pressure line, and the 98 per cent quality line, at which point the heat content, as read from the Mollier chart accompanying the Steam Tables, is 1175.5 B.T.U.

If the nozzles were perfect, the steam would expand in them at constant entropy to M. The heat content at M is 864 B.T.U. and the quality, 76.8 per cent. The available energy of the steam is $1175.5 - 864 = 311.5$ B.T.U., which would appear as kinetic energy of the jet emerging from a perfect nozzle. The nozzle is not perfect, however, and its losses, which are frictional in character, are returned as heat to the steam, which leaves the actual nozzle at some state B, on the one-pound pressure line, but with a higher heat content than at M. The first of the two steps in the transformation of heat energy into useful work has now been consummated. The available heat energy has been transformed into kinetic energy of the steam with whatever efficiency the nozzles were capable of.

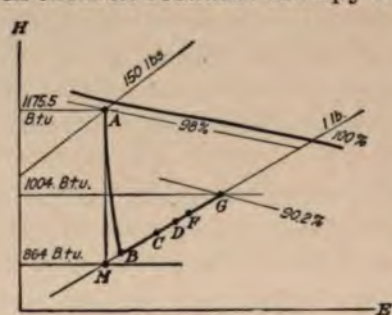

FIG. 118.

The kinetic energy of the jet is now to be absorbed, and in passing through the blade channels of the several velocity stages the stream is subjected to further frictional losses which result in the formation of heat that is returned to the steam as fast as generated. Thus in the first velocity stage, the steam is reheated to the state C; in the second stage to D; and in the third stage to F. After the steam emerges from the confinement of the blade passages of the

last velocity stage, it still has some "residual" velocity unclaimed by the blades. Furthermore the steam, or the particular pound of it whose history is being traced, spreads and momentarily fills the shell or chamber in which the rotor is revolving. The windage or rotation loss of the wheels cutting through the steam or whirling it about introduces a braking effect which contributes to the heat content of the steam before it passes on to the condenser. The combined reheating effect of the residual velocity and windage losses is represented by the vertical distance from F to G in Fig. 118. The steam enters the condenser at the state G.

Assume that the total reheating losses, including nozzle loss, blade friction, residual velocity, and windage, amount to 140 B.T.U. per pound of steam. The heat content at G is the sum of the originally unavailable heat, 864 B.T.U., and the waste, 140 B.T.U., or 1004 B.T.U. The utilized energy per pound of steam is $1175.5 - 1004. = 171.5$ B.T.U. The whole history of the steam while passing through the turbine is a continual depletion of the available energy by the successive losses encountered.

198. Heat Changes in a Multiple-pressure Stage Turbine.—The heat history of a pound of steam traversing a multiple-pressure stage turbine is much the same as that which occurs in the single stage turbine, with one important exception. In the single stage turbine, the heat that reverts to the steam as a result of frictional effects is irrevocably unavailable; in the multiple stage turbine, the heat that reverts to the steam as a result of frictional effects in any given stage becomes in a slight degree available in succeeding stages.

For illustration, assume a two-pressure stage turbine to operate between 150 pounds abs. pressure and 98 per cent quality, and 1 pound abs. exhaust pressure. Let the entire work of the turbine be divided nearly equally between the stages. The points A and M are located on the heat chart, illustrated in Fig. 119, and as the specifications are the same as for the illustrative example of the

single-pressure stage turbine, the total available energy of the turbines is the same, viz., 311.5 B.T.U. The duty of handling this available energy is to be divided approximately equally between two stages. Hence each stage will be assigned about $311.5 \div 2 = 155.7$ B.T.U. Theoretically, the heat content of the steam, as it leaves the first stage to enter the second, will be $1175.5 - 155.7 = 1019.8$ B.T.U., and the state point N will lie on the constant entropy line AM. The heat content and entropy being known, the point N is located. From the chart, the pressure corresponding, is found to be 17.2 pounds. The first stage nozzles will therefore expand the steam from 150 pounds pressure to 17.2, and the velocity thus created will be absorbed by an appropriate number of velocity stages. The second stage nozzles will then complete the expansion from 17.2 pounds to 1 pound, and the velocity generated will be absorbed by suitable velocity stages. The problem incidentally illustrates the disproportionality between the pressure range of an expansion, and the resulting available energy. In the first stage of the turbine the steam expands from 150 pounds pressure to 17.2 pounds (which is nearly atmospheric)—a range of 132.8 pounds. In the second stage the range is from 17.2 pounds to 1 pound or 16.2 pounds. The available energy in both cases is the same.

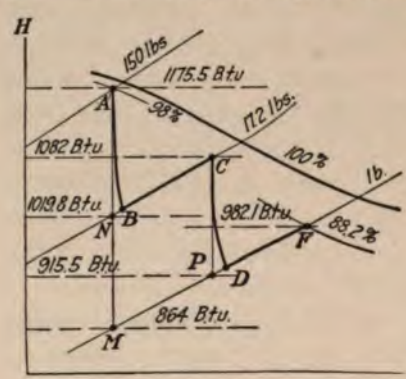

FIG. 119.

Assume the total reheating losses of each stage to amount to 40 per cent of its available energy. In the first stage the reheating losses $= 155.7 \times .40 = 62.3$ B.T.U. The heat content of the steam as it leaves the first stage, is 62.3 B.T.U. higher than the theoretical heat content at N.

The path of the state point of the steam, in the first stage, is *ABC*—complete expansion through the pressure range allotted, represented by *AB*, and constant pressurere heating along *BC*.

At entrance to the second stage, the steam pressure is 17.2 pounds, and the quality 92.6 per cent, represented by the point *C*, Fig. 119. In the second stage, the steam is to expand from this initial condition to 1 pound abs. From the chart, the available energy is found to be 166.5 B.T.U. The reheating losses total up to 40 per cent of the available energy $166.5 \times .40 = 56.6$ B.T.U. The heat content of the steam as it leaves the turbine—represented by point *F*—is 982.1 B.T.U. The final quality is 88.2 per cent. The state of the steam while passing through the entire turbine is represented by the path *ABCDF*, Fig. 119.

It will have been noted above that the available energy of the second stage is 166.5 B.T.U., whereas that of the first stage is 155.7 B.T.U., which was taken as one-half the total available energy between the initial and exhaust conditions specified for the turbine. Why should not the available energy be the same for the second stage as for the first? The answer is that some part of the available energy that became unavailable in the first stage is still available to the second. This fact can best be explained by reference to a sketch of the temperature-entropy diagram, Fig. 120. *A* is the initial state at 150 pounds pressure and 98 per cent quality. The total available energy, resulting from adiabatic expansion to 1 pound exhaust pressure, is 311.5 B.T.U., represented by area *KLAM*. The line *QN* is drawn so that $QLAN = KQNM = 155.7$ B.T.U. In the first stage, the actual expansion is represented by *ABC*; 40 per cent of the area *QLAN* is rendered unavailable above the pressure of 17.2 pounds. 62.3 B.T.U. are taken from above the 17.2 pound line, and placed below it, thus increasing the heat unavailable to the first stage by the area A_1NCC_1.

By virtue of the lower pressure range of the second stage, a part of the area A_1NCC_1, representing waste from the first stage, falls above the lower limiting condition of the second stage. Area $MNCP$ represents the portion of the 62.3 B.T.U. waste energy of the first stage that becomes available in the second. The value of $MNCP$ can be easily computed. The height of the strip A_1NCC_1 is 680°. The width is $62.3 \div 680 = .0916$ units of entropy. The height of the section $MNCP$ is 118°. Hence the area is $.0916 \times 118 = 10.8$ B.T.U. The total energy available in the second stage is $KQNM + MNCP = 155.7 + 10.8 = 166.5$

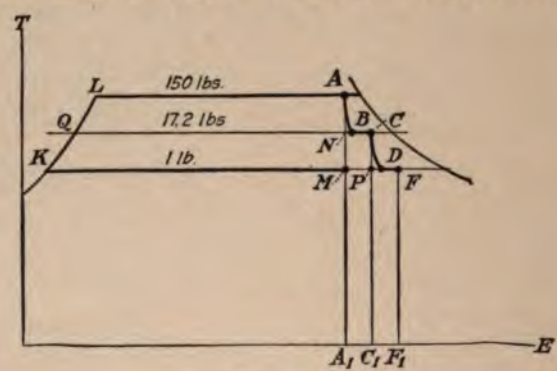

FIG. 120.

B.T.U. which agrees with the results of the computation from the Mollier chart.

On the Mollier chart the constant-pressure lines in the wet region are straight lines that diverge slightly as they proceed toward the right or drier region. The effect of the losses in the first stage of the two-stage turbine is to displace the state point from N (Fig. 119), its position if no first-stage losses occurred, to C, the actual state at entrance to the second stage. Whether the initial state of the steam with respect to the second stage is at N or C, the pressure range of the expansion is the same. But the available energy represented by CP is greater than that represented by NM.

199. The Reaction Turbine.—Whenever there is an acceleration of speed, as in the generation of velocity when a fluid passes through a nozzle, there is a backward push upon the nozzle itself. In the impulse turbine, the nozzle is held rigidly stationary in the frame, and consequently this reaction force does not become active in doing mechanical work. In the reaction turbine it is this backward push or thrust that is employed as the force element of the machine. An outstanding problem in the design of a reaction turbine is that of keeping the moving or traveling nozzle supplied with the steam or other working substance, under pressure, so that continuous expansion can occur, and a steady push be applied to the rotor.

In the Parsons type of steam turbine, which is the only example of a commercially built steam or gas turbine that embodies the reaction principle, the steam is thrown into the entrances of the traveling nozzles by stationary nozzles. The idea is illustrated in Fig. 121. C is the stationary steam chest supplied with steam from the boiler. D is a stationary nozzle whose function is to expand the steam from the initial pressure P_1 to some intermediate pressure P_2, such that the available energy of the expansion results in a velocity of the steam high enough to cause it to throw itself into the series of moving steam chests, E, on the rotor. The steam can then expand through the moving nozzles F, from the intermediate pressure P_2 to the final pressure P_3. The force of reaction pushes the rotor around. Since the steam is not fully expanded in the stationary nozzle, its elasticity or pressure will cause it to try to escape from the moving chests, E, through any opening offered. A reaction turbine must depend upon small clearance spaces, G, between stator and rotor, to prevent excessive loss by leakage.

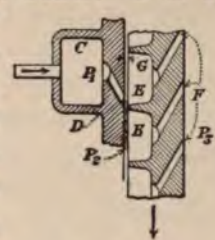

FIG. 121.

If the velocity generated in the stationary nozzle D

is higher than the speed at which the moving chests E are traveling, the moving element will experience a driving force when the jet from D strikes it. As a consequence the entire driving force acting will be made up partly of impulse and partly of reaction, and the turbine will operate upon a combination of the two principles.

The picture of Fig. 121 is intended to illustrate the idea of the reaction turbine, and does not at all represent the appearance of the actual steam elements of a Parsons turbine. In Fig. 121, the steam passages are given contours to bring out the idea that they are nozzles. In the actual machine the elements have more the appearance of blades, and are called blades, in fact; but their function is almost wholly that of nozzles.

200. The Steam Injector.—The steam injector is a familiar example of a kinetic machine, wherein the mechanical effect sought is that of a pump. The available heat energy of the steam is first transformed into kinetic energy. The momentum of the steam jet is then imparted to the water that is to be pumped, thus generating a high velocity of a column of water. The kinetic energy of the water jet is finally converted into pressure energy of sufficient head to enable the water to be delivered into the boiler or other high-pressure vessel into which it is to be pumped.

Fig. 122 represents the elements of an injector. Steam enters the chamber F through the pipe A, and is expanded by the nozzle N. The steam jet is thrown into the space E, where it commingles with water that enters around the jet from the supply pipe B. Usually the water enters because of a vacuum existing in E resulting from the condensation of the steam. From the mouth of the nozzle N to the point C, the steam imparts momentum and heat to the water. Or, more properly speaking, there is an equalization of the heat contents of the steam and water, because at C the water consists of the original water that entered through B, and the condensed steam. At the point C the water column has its maximum velocity. In

the divergent tube *D* the water is gradually slowed down, thus converting its kinetic energy into a pressure sufficient to impel it into the boiler through the pipe *G* against the resisting head. In starting the injector, when the steam is turned on and before the injector begins to function, the discharged and uncondensed steam finds a way of escape at *C* out through the overflow pipe *H*.

The first operation of the injector, that of transforming available heat energy of steam into velocity energy, is purely a steam nozzle function, and can be performed with very good efficiency. When the initial pressure and quality and the approximate amount of steam required

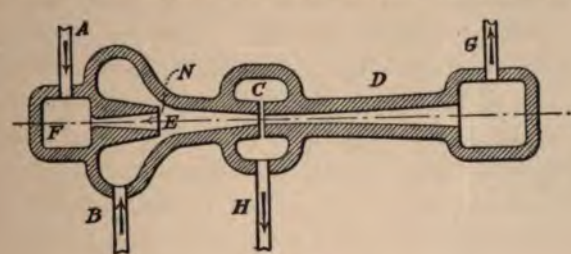

FIG. 122.

by the injector are known, the area at the throat can be determined. Further, if the back pressure on the nozzle in the space *E* can be predicted, then the relation of mouth to throat area can be determined. The length of nozzle is then set by the restriction that the total flare of both sides of the axis shall not exceed 12°. The steam nozzle can thus be satisfactorily designed for good efficiency by the nozzle laws already established.

The third operation is that of a reversed nozzle. The function of the tube *D* is gradually to slow down the jet of water and so convert kinetic energy into pressure. This is a process that has been found to be far less capable of efficient accomplishment than its reverse, viz., the generation of velocity from pressure. Turbulence of flow which is a frictional effect, resulting in the generation of heat

from work energy that should have been conserved during the operation, is far more difficult to prevent.

201. Impact.—Of all the three energy transitions performed in an injector, however, the second is by far the most wasteful. Into a relatively slowly moving column of water the jet of steam is hurled at terrific velocity—3000 ft. per sec. or more. Some of the momentum of the steam jet is imparted to the water, and some is destroyed without any velocity to show for it. But even if the entire momentum of the steam could be imparted to the water column, the operation would still be extremely wasteful of energy. For an example, assume 1 pound of steam per second emerging from the nozzle with a velocity of 3000 ft. per sec. Assume that for each pound of steam there is supplied to the injector 15 pounds of water, through the pipe B (Fig. 122), and that the water has a velocity of its own of 20 ft. per sec. just as it makes contact with the steam jet. Assume further that the original momenta of both the steam and water jets are completely conserved in the commingled stream at C. Then the velocity of the jet at C can be computed:

$$\frac{(1\times3000)+(15\times20)}{16}=206 \text{ ft. per sec.}$$

Now compare the kinetic energy possessed by the media before and after impact.

$$\text{Kinetic energy of steam jet} = \frac{1\times3000^2}{2g} = \frac{9{,}000{,}000}{2g}$$

$$\text{Kinetic energy of water before mixing} = \frac{15\times20^2}{2g} = \frac{6{,}000}{2g}$$

$$\text{Total before mixing} = \frac{9{,}006{,}000}{2g}$$

Kinetic energy of the combined jet at

$$C = \frac{16\times206^2}{2g} = \frac{679{,}000}{2g}$$

Only 7.5 per cent of the original kinetic energy has been conserved. The rest has been lost by impact.

Impact is, of course, another frictional effect,—an irreversible process. The 92.5 per cent of kinetic energy lost in the example above, appears as heat in the combined jet. Impact is an effect that the steam turbine designer endeavors to eliminate by curving the blades so that the steam enters tangentially to the surface. In fact, next to the proper proportioning of the area of section to the existent values of velocity and specific volume, the most important problem of all those that have to deal with the question of fluids in motion, is the avoidance of impact. The turbine's blade must receive the stream of steam, or air, or water, and change its direction in the gentlest possible manner. The pipe must change the direction of its column of contents without whirls or eddies, because if any kinetic energy is lost in impact, it will have to be made good by drawing upon the store of pressure energy. Impact is the essence of an irreversible process—one that is chaotic and disordered, always accompanied by splash and commotion.

202. Efficiency of the Injector.—With the second energy-transforming operation so remarkably low in efficiency as demonstrated in the preceding article, it must follow that the efficiency of the entire apparatus, as a pump, or heat engine, is exceedingly low. It is so low, in fact, that the injector cannot compete in economy with the very poorest of mechanical pumps. However, in the capacity in which it finds its widest use, viz., that of feeding boilers, the injector offers the advantage of perfectly conserving the heat supplied to it in the steam, returning practically 100 per cent to the boiler. But no matter how perfect a conserver of heat the injector is, it is a waster of available energy, and from a purely economical standpoint inferior to a feed-pump system in connection with a feed-water heater.

On the other hand, the remarkable simplicity of the injector, together with its small size and low cost, recom-

mend it for certain situations where economy is not the ruling consideration. Outside of the use of the injector for feeding boilers, it is frequently applied to the evacuation of spaces of water or air, or any other fluid. When so used, the instrument is generally called an ejector or aspirator, but the difference in name relates rather to the service performed than to any difference in principle. The motive fluid may be anything of which there is an available supply under pressure; and the substance handled may be anything that will flow. The cross-sectional areas, and contours of passages of an injector must always be designed with respect to the velocities and volumes of the media flowing.

The energy efficiency of the injector or ejector is much higher when the motive fluid is relatively heavy or dense in comparison with the substance to be pumped, as for example in the case of an ejector using water to partially evacuate a chamber of air.

Problems

1. Given a single-pressure stage steam turbine supplied with steam at 125 pounds abs. and 98.5 per cent quality. The exhaust pressure is 15 pounds abs. Assume the nozzle loss to be 6 per cent, the blade friction loss 18 per cent, the residual velocity loss 14 per cent, and the rotation loss 15 per cent, all expressed in terms of the available energy. Calculate the heat content and quality of the steam as it leaves the nozzle, as it leaves the blades, and as it leaves the turbine. Picture the heat changes on a sketch of the heat-entropy chart.

2. Given the turbine of Problem 1 with the same specifications, except that it is to be operated by air furnished at 400° F. Calculate the temperatures at the points indicated in Problem 1, and picture the heat changes by a sketch on the *TE* plane.

3. A two-pressure stage steam turbine takes steam at 160 pounds abs. and exhausts at 2 pounds abs.

a. Dividing the available energy of the steam equally between the stages, determine the pressure of the steam as its leaves the first stage and enters the second.

b. Assuming that 37 per cent of the available energy of the first stage appears in reheating of the steam due to losses, what is the heat content and quality of the steam as it enters the second stage, and what number of B.T.U. is utilized per pound of steam in the first stage?

c. What is the available energy of the steam for the second stage, considering the initial state of the steam as determined in *b*?

d. Assuming that 37 per cent of the available energy of the second stage appears in reheating, what is the heat content and quality of the steam as it leaves the turbine, and what number of B.T.U. is utilized per pound of steam in the second stage?

e. Calculate what part of the 37 per cent of the available energy of the first stage that was rendered unavailable to that stage by losses, becomes available to the second stage. Use sketch of temperature-entropy diagram in illustration.

4. In a given steam injector, the steam velocity is 3200 ft. per sec.; the velocity of the water with which it is commingling is 30 ft. per sec.; the injector handles 20 pounds of water per pound of steam. What is the loss of energy by impact in the chamber where steam and water meet, assuming complete conservation of momentum?

CHAPTER XVIII

THE KINETIC THEORY OF HEAT AND MISCELLANEOUS

203. The Foundations of Thermodynamics.—All the discussions and conclusions that constitute the study of Thermodynamics rest upon a relatively few basic facts. When formulated into statements, they are generally called laws. The fundamental data upon which the science of Thermodynamics is built may be summarized briefly as follows:

Boyle's Law, discovered by Boyle in 1662.

Charles' Law, discovered by Charles in 1787.

The analytical work of Carnot about 1824, which constitutes the essential elements of the Second Law.

The quantitative demonstration that heat is a form of energy, made by Joule when he first determined experimentally the mechanical equivalent of heat in 1843.

Joule's Law, based upon his experiment to show that when heat is added to a true gas, the increase of intrinsic energy consists of sensible heat only; there is no latent heat.

The experiments of Regnault in 1847, which furnished the first data on the properties of steam.

It is not to be understood that these facts or discoveries assumed at once upon their enunciation the outstanding character and importance that they have for us to-day. It sometimes required long periods of discussion, analysis, adjustment of existing theories, and settlement of corollary principles, before general acceptance was given the new contribution. Among the familiar names of the prominent

physicists who contributed to the science in this respect are Gay-Lussac and Dalton on the properties of gases, in the first decade of the last century; Clapeyron, who interpreted and further discussed Carnot's principle, and originated a general expression for the latent heat of vapors (about 1832); Clausius, who also studied especially the Second Law, being the author of the familiar statement "Heat cannot of itself, pass from a colder to a hotter body;" who was among the first to introduce the idea of entropy; and whose thermodynamic contributions occurred chiefly within the period 1850–1860; Rankine (1820–1872), who was really the founder of the modern science of Thermodynamics, and his was the first book on the subject; and Lord Kelvin (Sir William Thomson), who proposed a scale of absolute temperature independent of the properties of any substance, and also first presented the dynamical theory of heat 1851, in such a manner as to reconcile the work of theorists and experimenters on this subject, and lead to universal acceptance of the theory.

But the science of thermodynamics could never have become an actuality without the creative genius of inventors who fashioned their ideas into metal and built heat engines for practical purposes. Most of these men had some theoretical background upon which to erect their practical ideas. But in the invention of the steam engine, James Watt and his predecessors proceeded from direct physical conceptions derived from personal experience, without the aid of any formulated principles representing the thought and experience of others. The appearance of the first assuredly successful steam engine in 1765 preceded by a half century the active beginnings of the study of thermodynamics. The hot air external combustion engine of Stirling appeared in 1816, and that of Ericsson about 1833. Then succeeded a long period of study and unproductive invention on the internal-combustion engine culminating in Lenoir's engine of 1860, which was followed by the Otto and Langen free-piston

engine of 1867, the Brayton or Joule engine of 1870, the Otto engine of 1876 and finally the Diesel engine. The steam turbine as a successful possibility dates from the inventions of Parsons and De Laval, beginning about 1884.

204. Theories of Heat.—The science of thermodynamics is built upon observed facts of nature, and does not therefore depend upon the correctness of any theory of heat. It is sufficient to know only that heat is a form of energy. However, it is impossible to separate the study of thermodynamics from some speculation upon the nature of this all-important form of energy, and it is desirable to know what is the generally accepted theory of heat by men who have given the greatest thought and attention to the subject.

The earlier or "Caloric" theory supposed heat to be a substance, that had weight, and flowed from one containing substance to another like water, under proper conditions. It was Joule's experiment on the mechanical equivalent of heat that finally served to disprove the caloric theory. At the present time the kinetic theory of heat is universally accepted.

205. The Kinetic Theory of Heat.—The kinetic theory assumes that heat is a "mode of motion." As to just what this motion is, no one can say definitely. It can only be answered when the queston of the ultimate constitution of matter has been settled. But enough is known so that a fairly plausible picture of what goes on may be presented.

What is designated as matter or substance is not a solid homogeneous mass, but is in fact a structure composed of structural elements, the molecules, the smallest division into which a substance could be broken up without losing its identity. The molecules in turn are structures composed of atoms; and the atoms in their turn are structures made up of still smaller units called electrons. Presumably all electrons are alike. Electrons may arrange themselves in something more than a hundred different ways,

each different arrangement constituting a different kind of an atom, the fundamental unit of a different substance known as a chemical element. The atoms of different elements may unite in an infinite variety of combinations to form the molecules of all known substances. Matter, according to this theory, is therefore structural in nature, consisting of systems within systems.

Considering a molecular system, what are the forces that hold the atoms in place to form a definite configuration? It is believed that permanency is maintained by the average balance of two opposing forces; one an attraction which tends to concentrate the elements of the system; the other a centrifugal force resulting from orbital velocities, that tends to separate the constituent elements to a greater distance. The molecule is in fact a miniature solar system, although invisibly small in dimensions. The general relation between the sizes of the solidified masses of the sun and planets to the dimensions of the solar system is similar to those of the atomic masses as compared with molecular dimensions. Furthermore, an entire molecular system, while its component parts are involved in mutual relative motions, may be participating as a unit, in motions relative to surrounding molecular systems. While the magnitudes of the dimensions of molecules are so small as to be far below the reach of the most powerful microscope, the velocities of their motions, and the corresponding kinetic energies are enormous.

Heat is the kinetic energy represented by this intermolecular motion. Quantity of heat is determined by the aggregate of the mass of molecules involved, together with the intensity of the energy represented by $\frac{V^2}{2g}$, where V is the average linear velocity of the molecules. Temperature is representative only of the intensity of the kinetic energy viz., $\frac{V^2}{2g}$.

Temperature, as a sense perception, is the sting that

is felt when the skin is impinged upon by the molecules of a hot substance when examined by touch.

The proximity or disgregation of the molecules of a substance determine its state as a solid, a liquid, or a vapor. In a solid state, the molecules are relatively near each other and the force of attraction upon one another is large. When a solid is heated, the velocities of the molecules increase, until a point is reached when the centrifugal force is sufficient to begin to break up molecular systems, and the particles wander about in the mass. As heating continues, no further increase of velocity takes place, and consequently no further rise in temperature occurs, until uniformity of condition throughout the mass has been reached. The solid changes to a liquid and the latent heat of the liquid is accounted for in the breaking up of the molecular systems against their mutual attractions. Latent heat is the potential energy of separation of attracting masses.

With further application of heat, further increase of velocity takes place until a condition is reached when the force of separation is sufficient to result in some of the molecules being thrown out of the mass of liquid entirely and vaporization begins. The wide displacement of the particles means a large increase in potential energy of separation of the molecules, and a consequent large supply of latent heat. The separation is so extensive that the volume of the entire mass is greatly increased, necessitating a considerable amount of energy to enable the substance to make room for itself against the environing pressure. This is the external work part of the latent heat of vaporization.

It thus appears that, after all, heat is a form of mechanical energy, and the difference between heat and what we call mechanical work is that the latter is a function of forces and motions of visible or tangible masses of matter, whereas the former is the aggregate of motions of what are to us the infinitesimal constituents of matter. Whether heat is the energy of motion of molecules alone, or whether

atomic and electronic motions within the molecule participate in the effect, does not alter the general conception inasmuch as it is the aggregate of the kinetic effect that is perceived as heat.

206. Combustion.—Combustion, according to the kinetic theory of heat, is a transformation of potential energy of molecular masses into kinetic energy. When a weight is held separated from the mass of the earth, there is a potential energy of position. If the props that hold the mass from the earth are suddenly removed, the weight will acquire motion toward the earth. Potential energy will be gradually transformed into kinetic energy. The kinetic energy of the weight will not persist long in that state, however. Soon the motion is stopped by impact. The energy passes out of the realm represented by motions and masses discernible to us into the realm of molecular motions and masses invisible to us except by inference.

Suppose that the weight in the illustration used above could be held out at a constant distance from the center of the earth, while the earth shriveled in size to only a very small fraction of its original diameter. Then let the weight be released, as before. It will fall towards the earth, but not straight toward it. Its motion will be a composite of the previously possessed motion of rotation with the earth, and the induced continually accelerated motion due to attraction. The weight may not strike the earth, but may pass on by it with a decelerated motion following its point of nearest approach. Under the influence of the forces acting upon the weight—its mutual attraction with the earth, and the centrifugal force due to velocity—it may continue to oscillate indefinitely, assuming a definite orbit of travel around the earth. Thus the original potential energy of position may be conceived as transformed into a kinetic energy of the weight, that persists indefinitely.

Combustion is the falling together of atoms, urged by their attraction for each other. Under ordinary circumstances the atoms of carbon in coal and those of the oxygen

of the air are not sufficiently near each other to enable their attraction to pull them together. But with sufficient heat, which means increase of velocity of molecules, some of them may be thrown near enough together to be seized by their mutual attraction and fall together like the weight and the earth, with a motion perhaps somewhat similar. The activity is imparted to adjacent particles, and their increased motion brings other pairs of molecules within seizing distance, and so the process continues, and combustion takes place.

The energy transformed in combustion is enormous. When the atoms of 1 pound of carbon and $2\frac{2}{3}$ pounds of oxygen fall together 14,500 B.T.U. or over 11,000,000 foot-pounds of energy are represented, which is more than the equivalent of the kinetic energy of a ton weight falling one mile.

207. Vapors and Specific Heat.—In the vapor state the molecules of a substance are widely separated relative to their configuration in the liquid and solid states. The attraction between molecules, although not zero, is very weak, and the motions of the particles are not much influenced by the presence of each other. The path, or probably it is more accurate to say the orbit, of any individual molecule becomes more nearly a straight line, and may be of considerable length before leading sufficiently near another molecule to be bent aside or turned about in direction. The bonds of the mutual attraction of the molecules are too weak to maintain the substance as a coherent mass. The velocity of the particles carries them farther and farther in every direction, unless the vapor is confined within a chamber, in which case the effort of the molecules to escape produces an elastic pressure upon the confining walls.

When a vapor is superheated, the molecular velocity is increased, and the interattraction of molecules is further weakened. Hence some of the heat of superheat is really latent heat of separation, instead of energy appearing

as increased velocity, which is temperature. With further superheat, the molecular attraction is still further weakened until it is scarcely discernible. The vapor approaches the characteristics of a true gas. This explains the variation in the specific heat of superheated steam. After steam has been superheated a few degrees the specific heat becomes considerably less than it was in the near vicinity of the saturation temperature, where the latent heat effect is still felt. For a similar reason the specific heat is higher for the higher pressures; the molecules are compressed into closer proximity to one another, in which condition the attracting forces are more powerful.

Again, as the temperature of a vapor becomes higher and higher, it has been shown that its specific heat increases. The increase is explained by the approach toward dissociation temperatures. Dissociation is a separation of the atoms that compose a molecule, against the mutual atomic attractions, resulting in a resolution of a substance into its chemical elements. The heat applied to effect dissociation is latent heat like the heat of vaporization. While the temperature of dissociation is definite for any particular gas, it is probable that the influence of the approaching change of state begins long before that temperature is reached, thus introducing a latent heat effect into the specific heat, increasing with the temperature.

208. Perfect Gas.—In a perfect gas—a hypothetical substance that would conform exactly to the laws of Boyle, Charles, and Joule—the attraction of the molecules for each other would be zero. Hence the paths of the molecules in their motions would be rectilinear. The molecules would be perfectly elastic, so that upon collision with the walls of a confining vessel, or with one another, the flying particles would rebound with their original velocity.

The impulse of the blows of the molecules upon the walls of the containing vessel is evidenced in the elastic pressure of the gas. Imagine a rectangular box, of dimensions x, y, and z, containing a perfect gas. Let the total

number of molecules be n, Let v represent the average velocity of the molecules. Each time a molecule impinges upon a wall the momentum produced will be $2mv$, if m is used to designate the mass of the molecule. If two faces of the box are considered, of which x and y are the dimensions, the molecule may be imagined to bounce back and forth from face to face, traveling a distance z between. The number of times the particle will impinge upon either face per second will be $\frac{v}{2z}$. The total force produced by the one molecule is

$$2mv \times \frac{v}{2z} = \frac{mv^2}{z}$$

By the law of averages, one-third of the total number of molecules, n, in the box, will be traveling in the direction z. The total force exerted upon the face xy will be

$$\frac{n}{3} \times \frac{mv^2}{z} \text{ or } \frac{2n}{3z}\left(\frac{mv^2}{2}\right)$$

Let P = the force or pressure per unit of area.

Then $$P = \frac{2n}{3zxy}\left(\frac{mv^2}{2}\right)$$

But zxy = the volume, V

Hence $$PV = \frac{2n}{3}\frac{mv^2}{2}$$

$\frac{mv^2}{2}$ represents the intensity of the kinetic energy of the molecular motion. It is proportional to the absolute temperature T, of the gas. Also the weight $\overline{W}$ of the gas in the box is proportional to the number of molecules contained. Hence,

$$PV = \overline{W} \times \text{constant} \times T, \text{ or } PV = \overline{W}RT,$$

which is the characteristic equation of a perfect gas. The kinetic theory of heat therefore conforms to the facts of thermodynamics as determined experimentally

by Boyle, Charles, and Joule, since the above equation, established by them, is found to be a necessary result of the theory.

209. The Equation of an Imperfect Gas.—In an actual gas the molecules do exert an influence upon each other, and they also occupy some space within the volume in which their activities take place. A perfect gas requires that the molecules should occupy no space. These deviations of an actual gas from the properties of a perfect gas necessitate a modification of the characteristic equation of the perfect gas, in order to make it represent accurately the relation of P, V, and T.

Since, in an actual gas, the molecules themselves occupy some of the space in which their activities occur, the clear volume is smaller than the nominal volume V by some amount b. Also since the molecules exert an attraction upon each other, the pressure of an actual gas for a given temperature and volume is less than it would be if it were a perfect gas. Or, stated in another way, if the imperfection of molecular attraction could be removed, the pressure of the gas would be increased by some amount that is an inverse function of the distance of separation of the molecules, and hence of the volumes. Van der Waals gives this correction as $\frac{a}{V^2}$, where V represents specific volume.

The equation of the perfect gas is

$$PV = RT$$

Introducing the corrections to P and V noted above, the equation of an imperfect gas may be written:

$$\left(P + \frac{a_2}{V}\right)(V - b) = RT$$

where P = pressure of the imperfect gas per unit area;
V = specific volume;
T = absolute temperature;
R is the constant of the gas if perfect, and a and b are

constants whose values can be determined for any gas if two sets of values of P, V, and T are known.

The values of the terms a and b, although considered constants, are in reality variables depending upon the shape of the volume in which the gas is confined, and the weight of gas considered.

210. The Joule-Thomson Effect.—Our conception of the point of absolute zero of temperature is derived from the behavior of gases under imposed changes of temperature. When a gas is cooled, for example at constant pressure, a reduction in volume occurs. If cooling could be continued indefinitely, and the volume continued to decrease at the same rate observed within the natural temperature range of our experience, a temperature could be found by exterpolation of the curve, at which the volume would be zero (Art. 79). This temperature is called the point of absolute zero.

If the absolute zero were determined by using different actual gases, there would not be perfect agreement. None of the gases follow exactly Boyle's and Charles' Law, and each is slightly different from the others in its deviation from the behavior of a perfect gas. It was proposed by Thomson that a more nearly exact determination of the absolute zero could be made by ascertaining the deviation of an actual gas from a perfect gas, by direct experiment, and then correcting the curve of the actual gas by the results. In cooperation with Joule, Thomson performed the famous " porous plug " experiments, and determined what is called the " Joule-Thomson effect."

The porous plug may be thought of as a tube stuffed with cotton or other fibrous material, and when a gas is throttled through it, kinetic energy is reconverted into heat almost as soon as it is formed, and the gas emerges from the tube with practically no velocity to be transformed back into heat in the exhaust chamber.

The results of the experiments were announced in 1862.

The Joule-Thomson effect is expressed as a difference between the temperature after throttling and before, and is a measure of the imperfection of the gas, because with a perfect gas, there should be no change of temperature. For air,

$$\text{Fall in temperature} = \frac{P_1 - P_2}{4}\left(\frac{289}{T_1}\right)^{2\cdot 1}$$

where P_1 = pressure before throttling, in atmospheres;
P_2 = pressure after throttling, in atmospheres;
and T_1 = temperature before throttling, degrees, C. absolute.

At ordinary temperatures, the cooling effect is about $\frac{1}{4}$° C. for each atmosphere of pressure drop. But it is seen by the relation of T_1 in the formula, that the drop in temperature is greatly magnified as the initial temperature of the operation is lowered.

At ordinary temperatures nearly all gases, like air, exhibit a fall in temperature resulting from throttling. Hydrogen is an exception, as a slight rise of temperature accompanies throttling. But if hydrogen is cooled to a sufficiently low temperature before throttling it then behaves like other gases. It appears that there is for every gas a point of inversion of the Joule-Thomson effect, occurring at a temperature of about 6.75 times the temperature at the critical point. Our ordinary temperature region is more than 6.75 times the critical temperature of hydrogen.

211. Critical Point of Gases.—For every gas there is a certain temperature above which it is impossible to liquefy the gas by pressure alone. The critical temperature of steam has already been mentioned in Art. 21.

In Fig. 123, a temperature-entropy diagram is represented. The liquid and saturation lines, if carried to a sufficiently high temperature, will merge at the critical

[1] From "Liquid Air," by T. O. Sloane, Norman W. Henley Co. 1899.

temperature T_c. P_1, P_2, and P_3 represent constant-pressure lines.

Take some point M, on the constant-pressure line P_1. If the pressure is increased without change of temperature, the state point follows the path MN, and at N, at some pressure between P_2 and P_3, the gas will begin to liquefy. But if the initial point be taken at Q, on the constant-pressure line P_1, it is clear that liquefaction by increase of pressure alone, without lowering of temperature, cannot be accomplished.

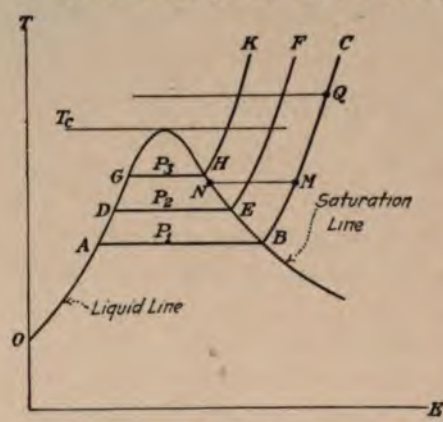

FIG. 123.

212. The Liquefaction of Gases.—The principle of the Joule-Thomson effect is employed in the liquefaction of gases by mechanical means. The first machine of the kind was developed by Linde. The essential idea of the Linde machine is represented in Fig. 124. B and D represent the cylinders of a two-stage air compressor. Atmospheric air or other gas entering at A is compressed to about 16 atmospheres in B. It is then delivered through a connecting pipe to the second cylinder D, where the pressure is raised to nearly 200 atmospheres. On the way from B to D the stream of air described is joined at C by another stream whose source will be made clear as the description proceeds. The air discharged from the second cylinder is delivered through a cooler E, and some sort of a drier, not shown. The compressed air, cooled and dried, enters the space F, and passes downward to the valve G, through which it is throttled to approximately 16 atmospheres, passing into the concentric annular space, H. K is another

throttle valve through which a portion of the air in space H escapes into the second annular space L. The remainder of the air in H returns to the high-pressure cylinder of the compressor, joining the new supply of air at C.

Cooling of the air is accomplished by taking advantage of the Joule-Thomson effect by the double throttling at G and K. With the pressures mentioned, and assuming the air in F to be at 32° F. there would be a cooling of about 93° F. by the first throttling and about 10° F. by the second, which is far from a sufficient drop to result in liquefaction.

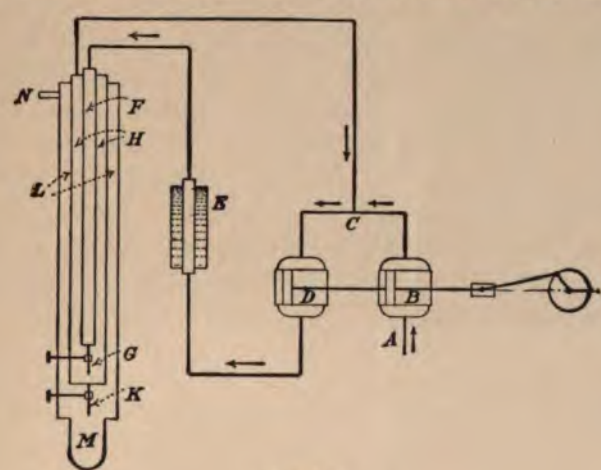

FIG. 124.

But most of the air cooled by the first throttling passes back through H, thus jacketing and cooling the primary air in F, so that as the operation progresses the temperature of the air coming to the valve G becomes lower and lower, and ultimately will fall so low that a part of the air that is expanded twice will be liquefied, and collect in the vessel M. What is not liquefied will pass up through the second annular space, jacketing H, thus assisting in cooling the primary air in F, and finally escaping to the atmosphere at N. The process is seen to be accumulative and once a constancy of temperature has been established in the

apparatus, it will deliver liquid air continuously. Moisture in the air is fatal to successful operation, as the ice will cause trouble at the throttle valves.

In the actual machine, the refrigerator is constructed of three concentric coils, instead of the three cylinders illustrated in the figure. By means of the coils, a sufficiently large surface is furnished to enable the transmission of the necessary amount of heat to take place.

At atmospheric pressure, air liquefies at $-312°$ F. Its critical temperature is $-220°$ F., the pressure corresponding to which is 573 pounds per sq. in.

Hydrogen liquefies (at atmospheric pressure) at a temperature of about $-422°$ F. By the reduction of pressure on liquid hydrogen, solidification takes place at about $-429°$ F., and evaporation from the solid produces a temperature of $-436°$ F., or 23.5° F. absolute.

The nearest approach to the absolute zero of temperature that has been attained is in the liquefaction of the gas helium which was first accomplished in 1908. The boiling-point of helium is 7.2° F.

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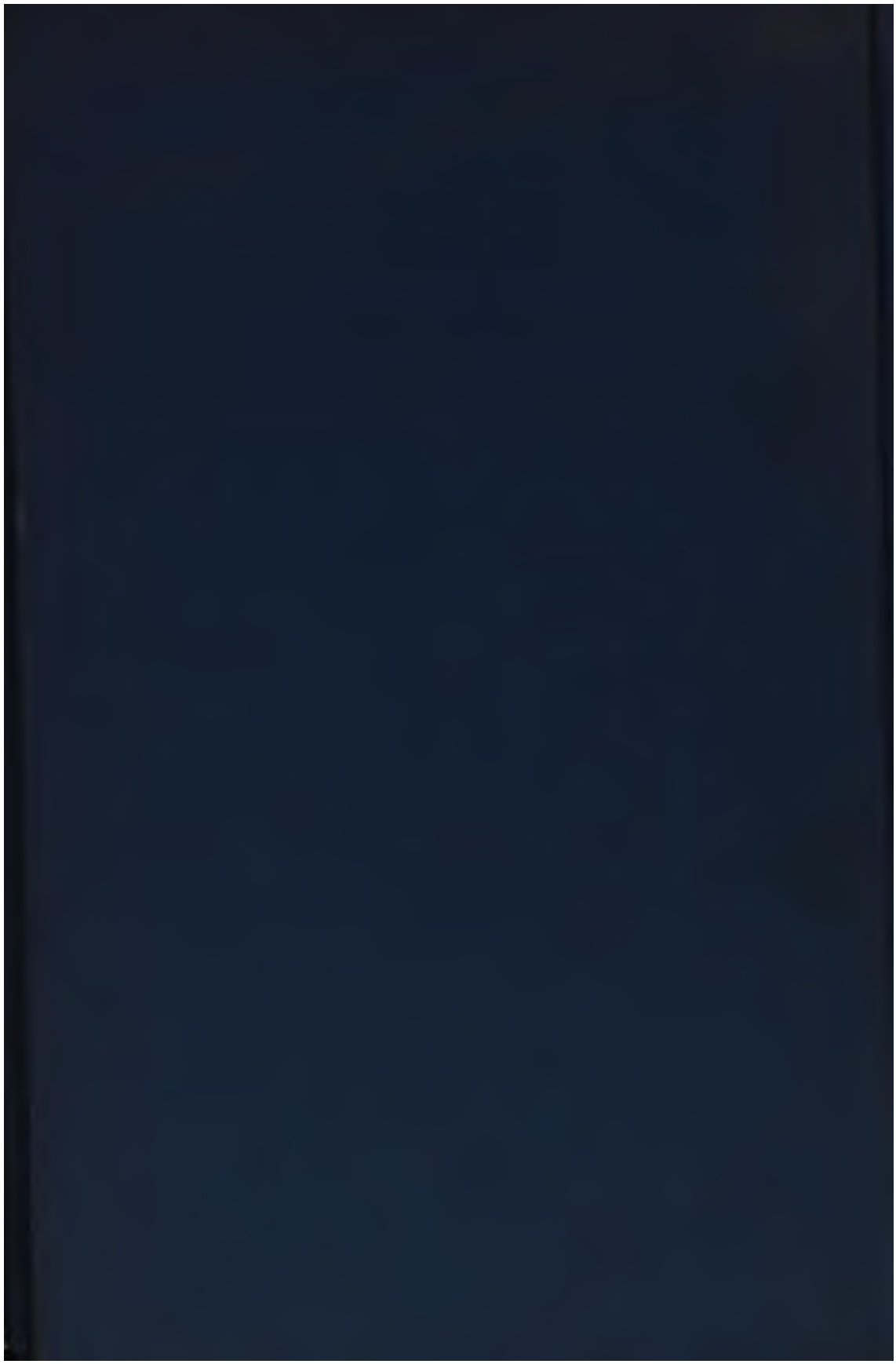
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